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Puiu

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(45) **Date of Patent: Oct. 16, 2007**

(54) **ACTIVE TORQUE COUPLING WITH HYDRAULICALLY-ACTUATED ROTARY CLUTCH ACTUATOR**

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(73) Assignee: **Magna Powertrain USA, Inc.**, Troy, MI (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 272 days.

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(22) Filed: **Jul. 8, 2005**

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Related U.S. Application Data

(63) Continuation-in-part of application No. 10/771,664, filed on Feb. 4, 2004, now Pat. No. 6,945,374.

(51) **Int. Cl.**
F16D 25/0638 (2006.01)
B60K 17/35 (2006.01)

(52) **U.S. Cl.** .... **192/35**; 192/85 AA; 192/93 A; 180/249

(58) **Field of Classification Search** .... 192/35, 192/70.23, 85 AA, 93 A; 180/249, 250; 475/86

See application file for complete search history.

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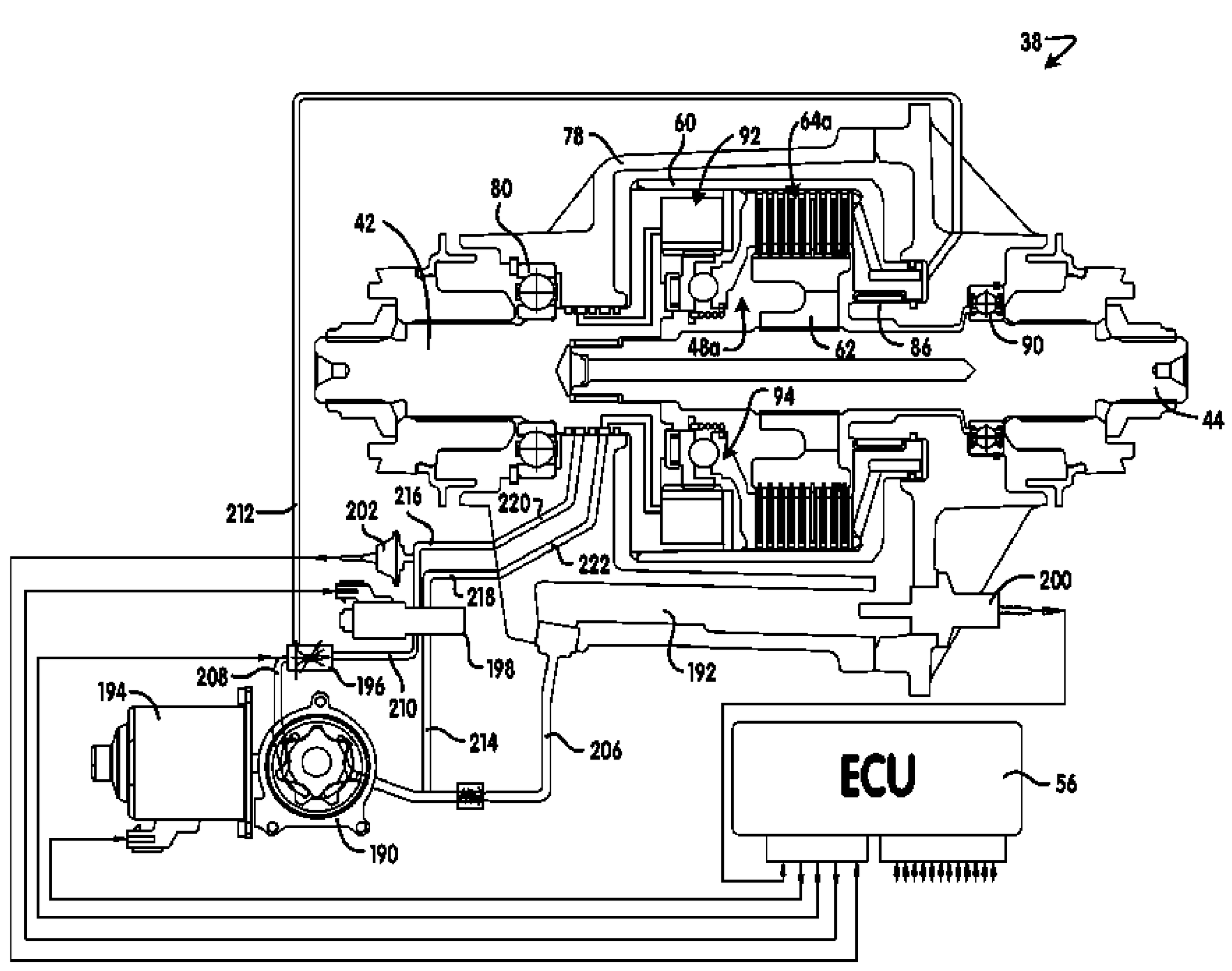
Primary Examiner—Richard M Lorence
(74) *Attorney, Agent, or Firm*—Harness, Dickey & Pierce, P.L.C.

(57) **ABSTRACT**

A torque transfer mechanism having a multi-plate friction clutch connecting a pair of rotary members and an electro-hydraulic clutch system for controlling engagement of the friction clutch. The electrohydraulic clutch system can include an actuator with first and second components with a plurality of actuation chambers that are employed to control the position of the second component relative to the first component. Relatively high pressure fluid may be directed through or around one or more bypassed portions of the actuation chambers when relatively low compressive clutch engagement forces are to be exerted on the friction clutch. The bypassed portions of the actuation chambers may be sequentially or progressively operated based on a position of the second component relative to the first component to increase an amount of force that is applied by actuator.

36 Claims, 20 Drawing Sheets



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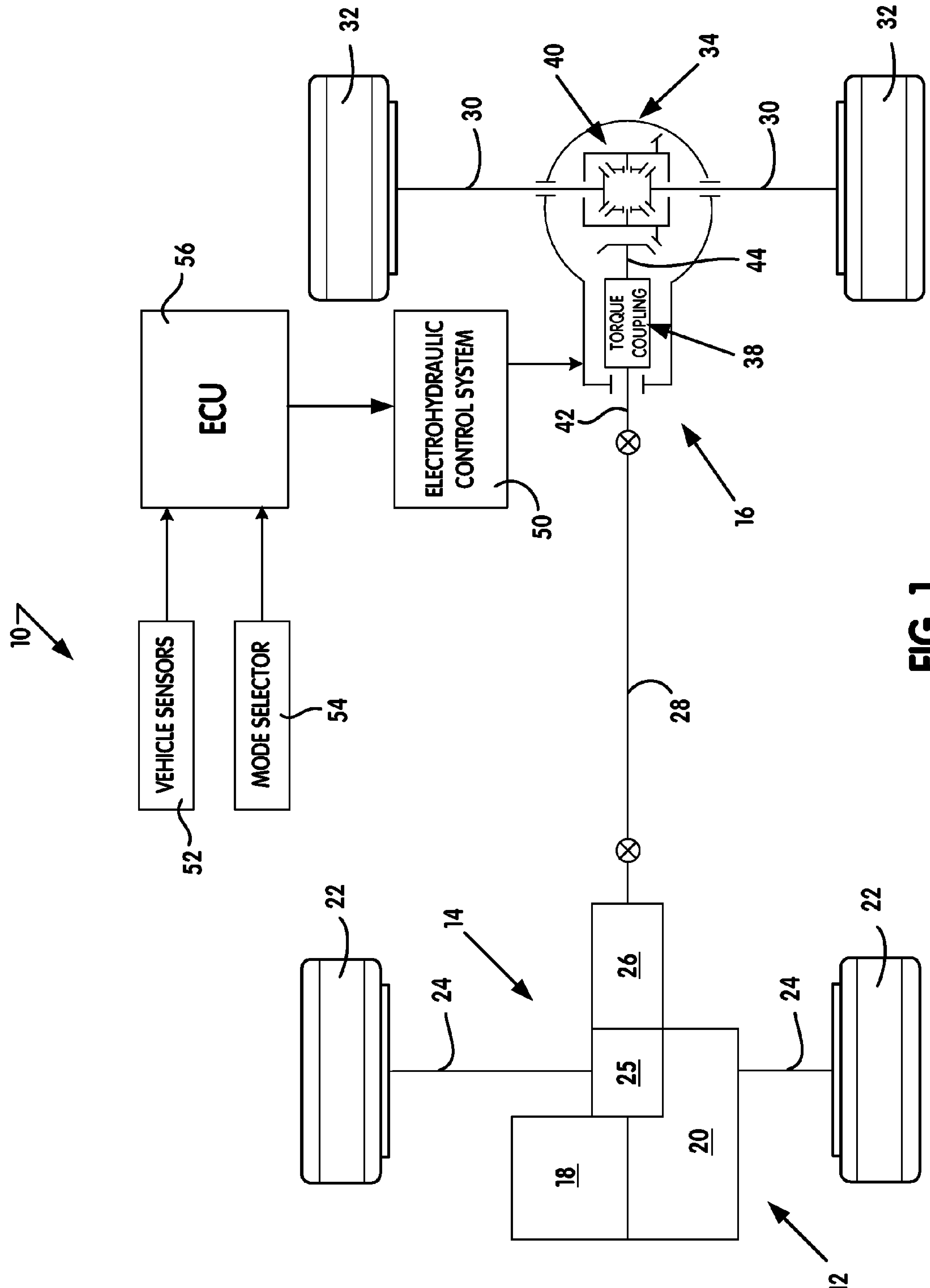
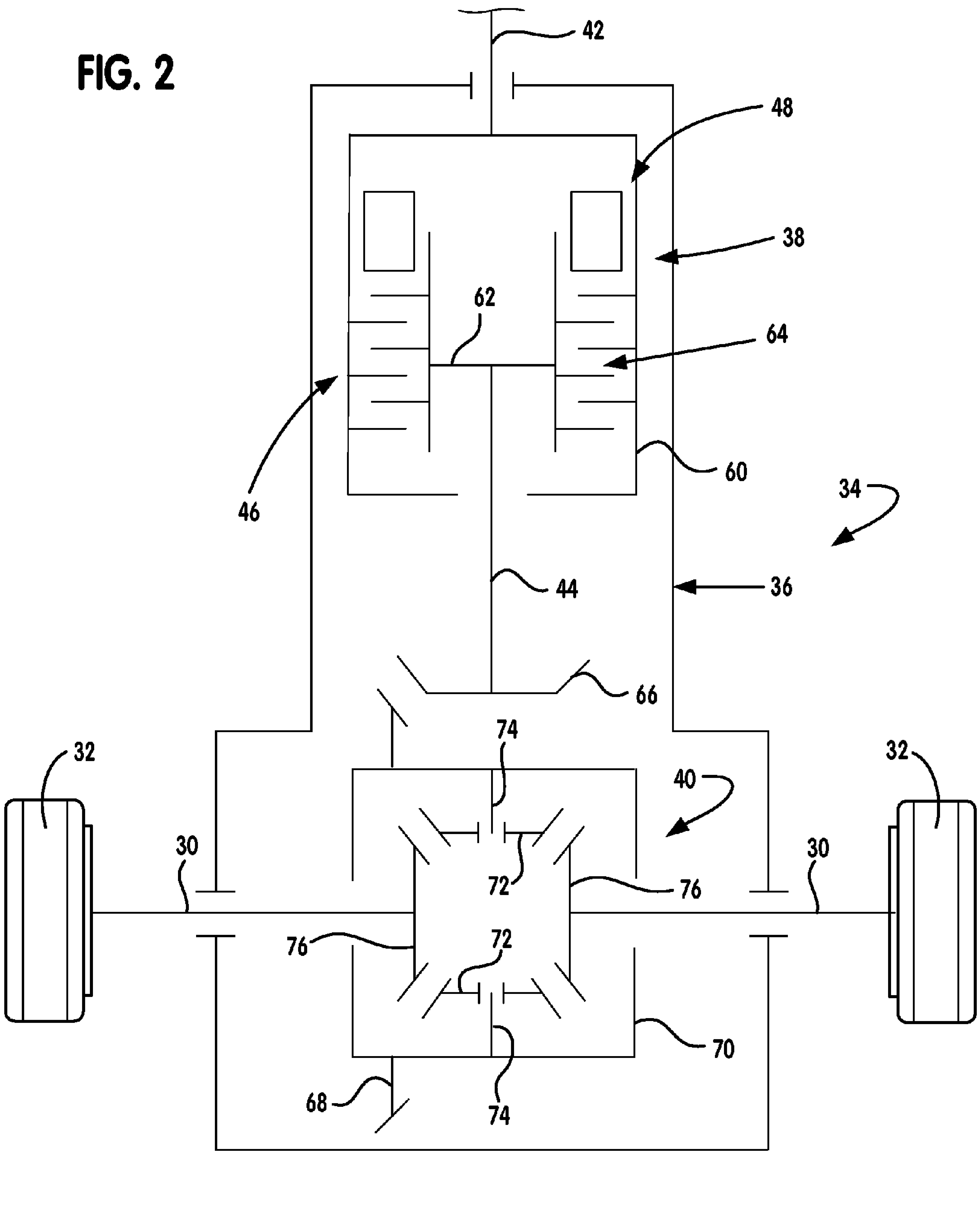
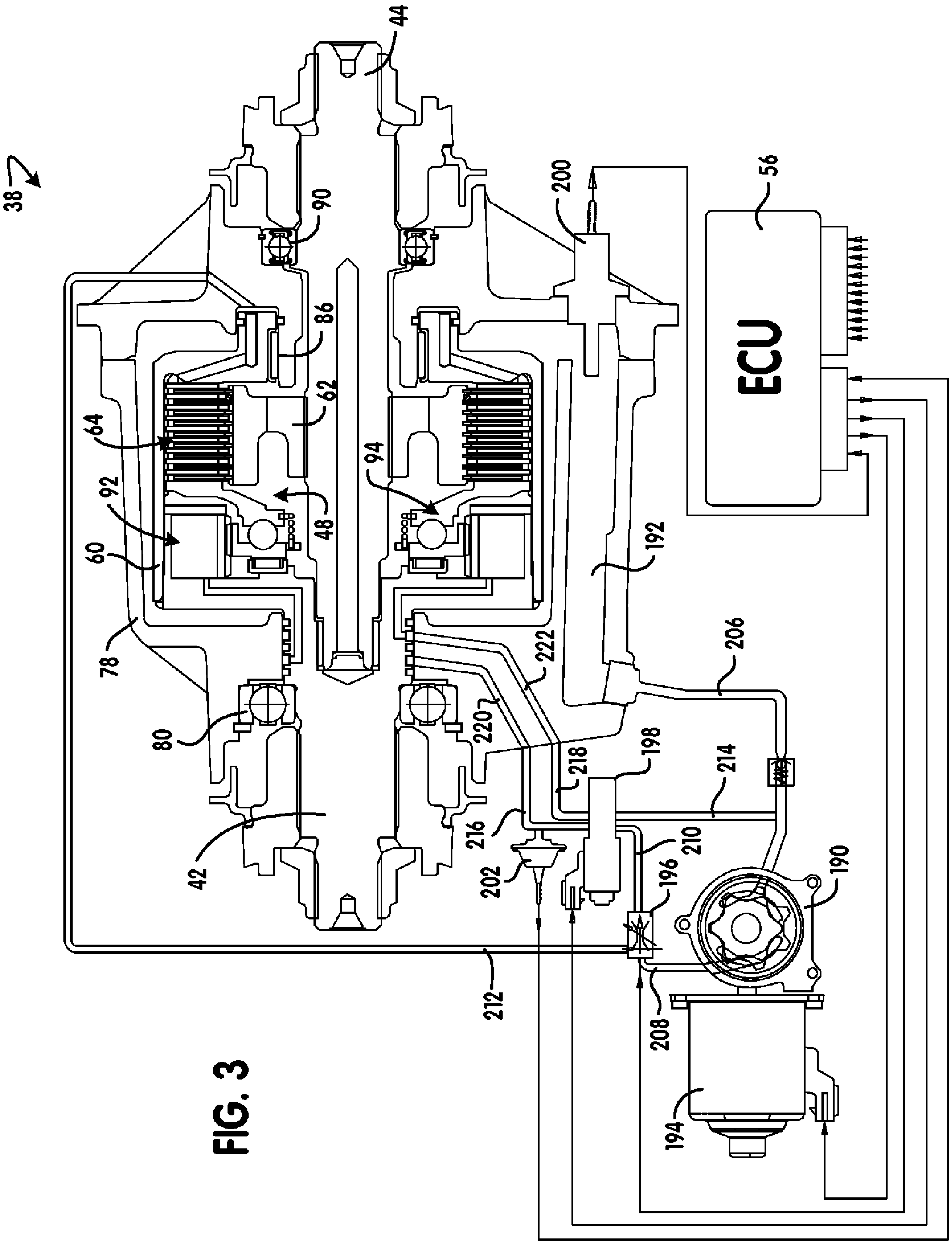

ECU
ELECTROHYDRAULIC CONTROL SYSTEM
TORQUE COUPLING
VEHICLE SENSORS
MODE SELECTOR

FIG. 1

FIG. 2

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44
90
86
62
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92
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48
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78
80
42
56
ECU
200
192
206
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202
210
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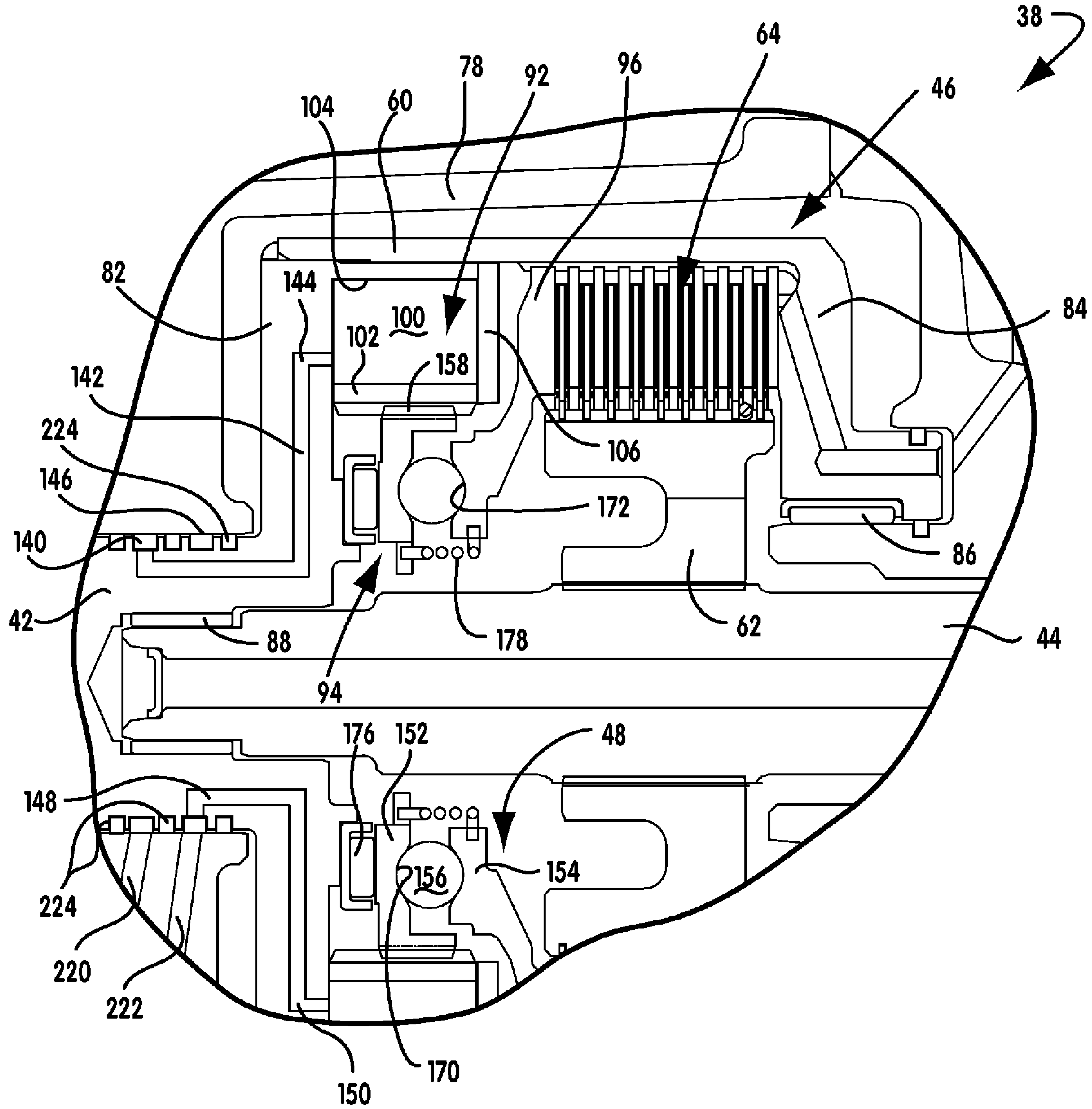
190
212
208
194

FIG. 3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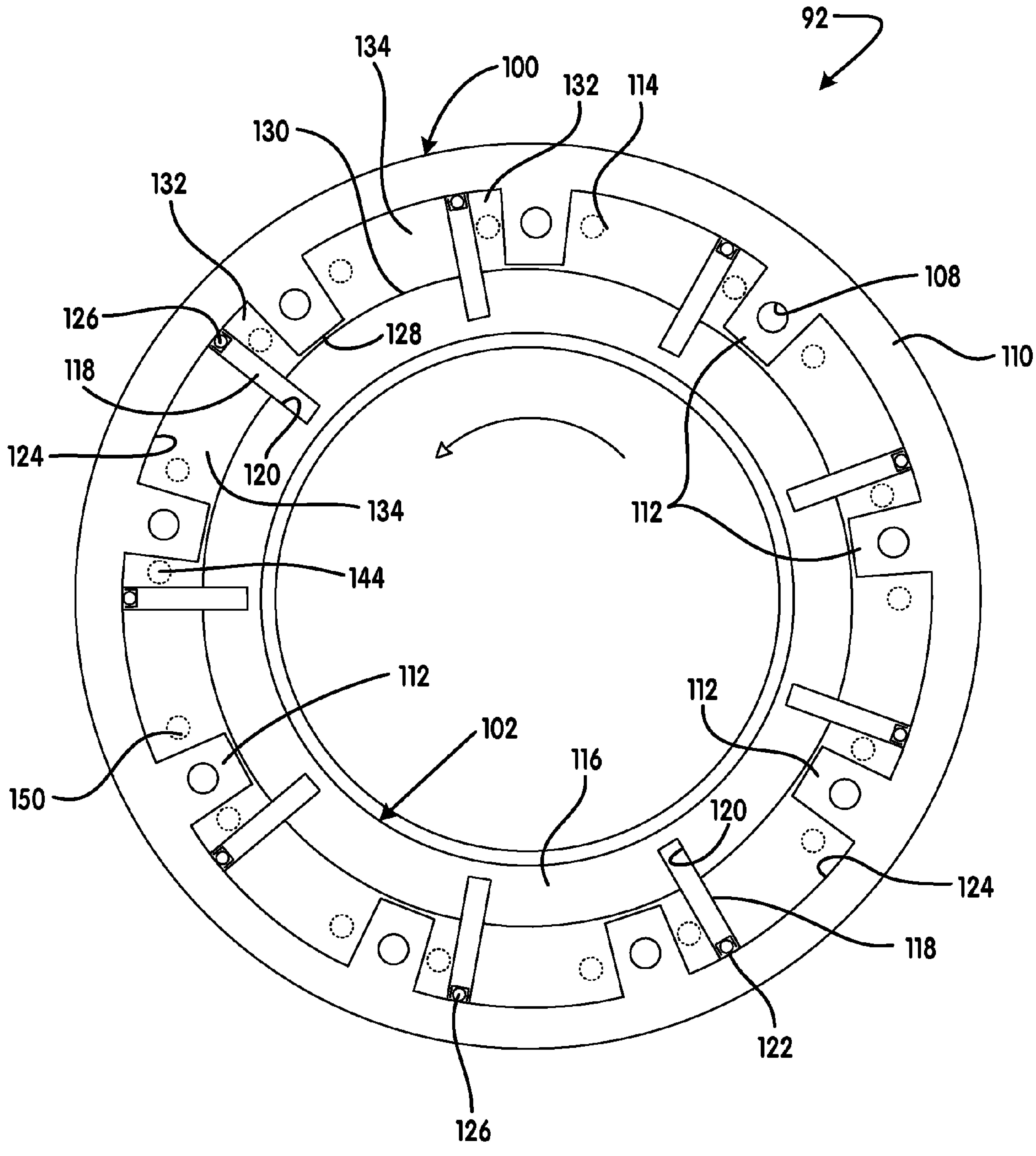
88
178
62
44
94
176
152
48
148
154
156
224
220
222
150
170

FIG. 3A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100
132
114
130
132
108
126
128
110
118
124
120
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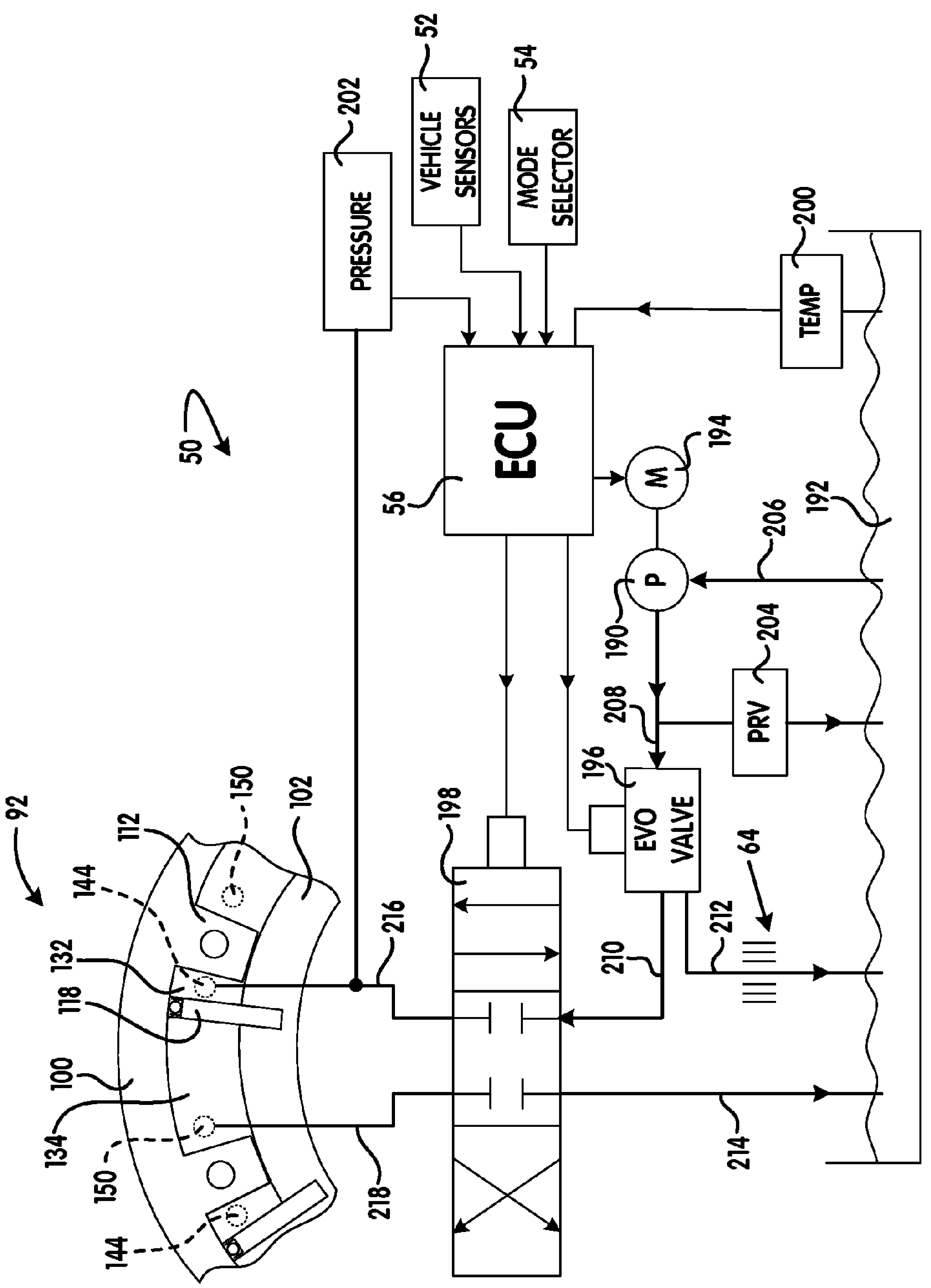
144
112
112
102
116
150
120
124
118
122
126

FIG. 4



PRESSURE
VEHICLE SENSORS
MODE SELECTOR
TEMP
ECU
M
P
PRV
EVO VALVE
202
52
54
200
50
56
194
192
206
190
204
208
196
198
92
150
102
112
144
132
118
100
134
216
210
212
64
214
218

FIG. 5

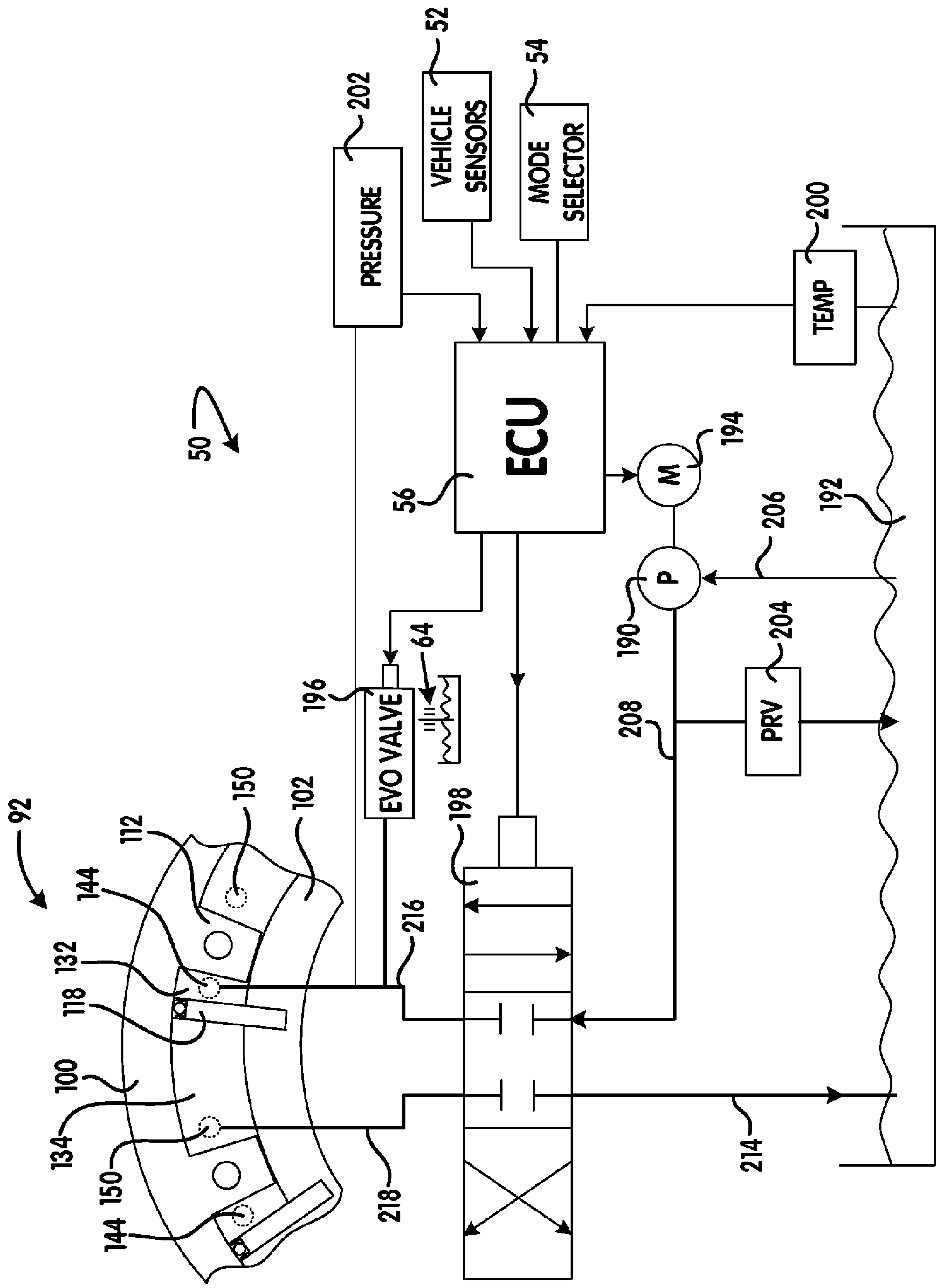
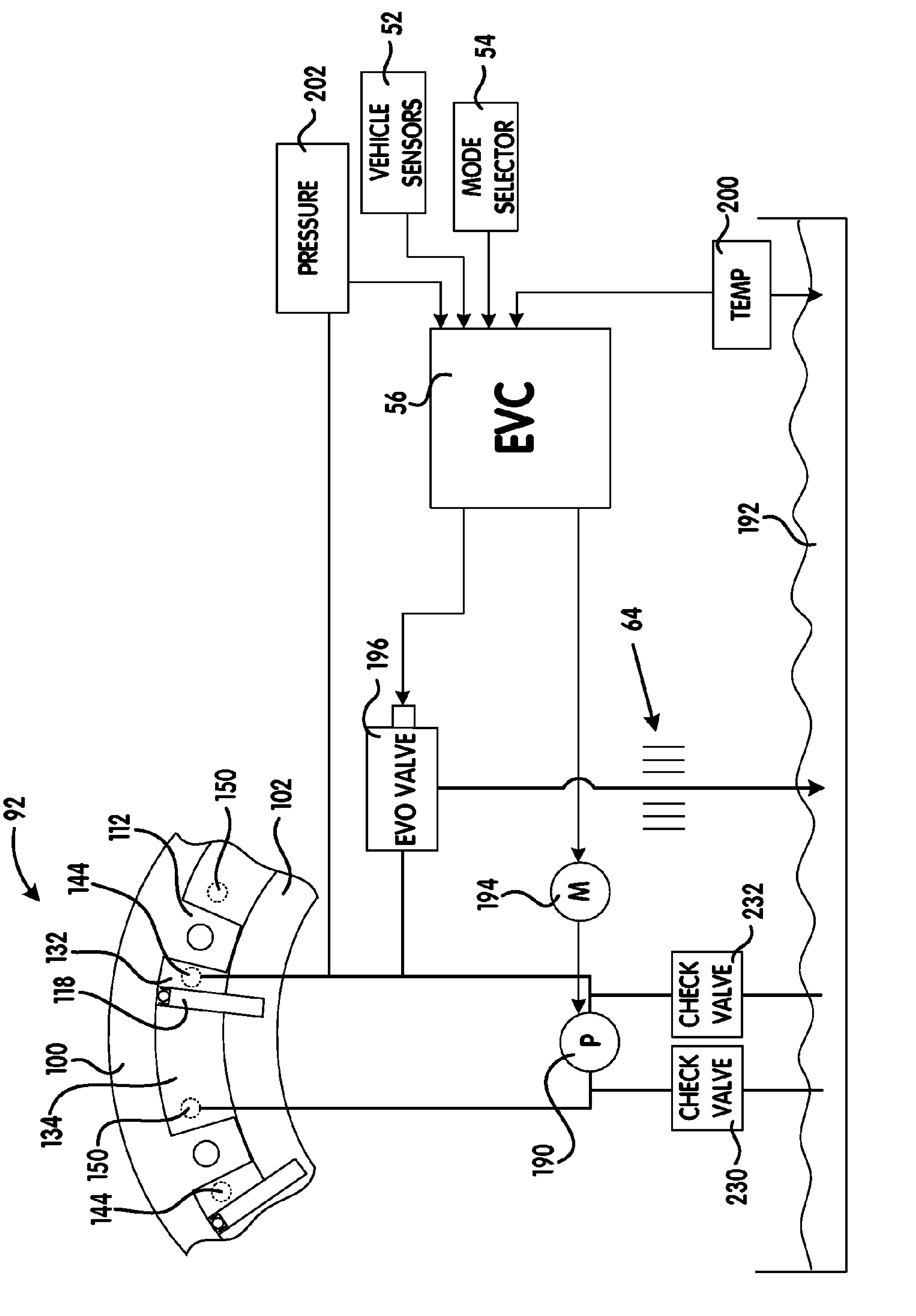

PRESSURE
VEHICLE SENSORS
MODE SELECTOR
TEMP
ECU
M
P
PRV
EVO VALVE
202
52
54
200
50
56
194
206
192
190
204
208
196
64
198
92
150
102
112
144
132
118
216
100
134
150
144
218
214

FIG. 6



PRESSURE
202
VEHICLE SENSORS
52
MODE SELECTOR
54
TEMP
200
EVC
56
EVO VALVE
196
64
192
92
150
102
112
144
132
118
100
134
150
144
M
194
P
190
CHECK VALVE
232
CHECK VALVE
230

FIG. 7

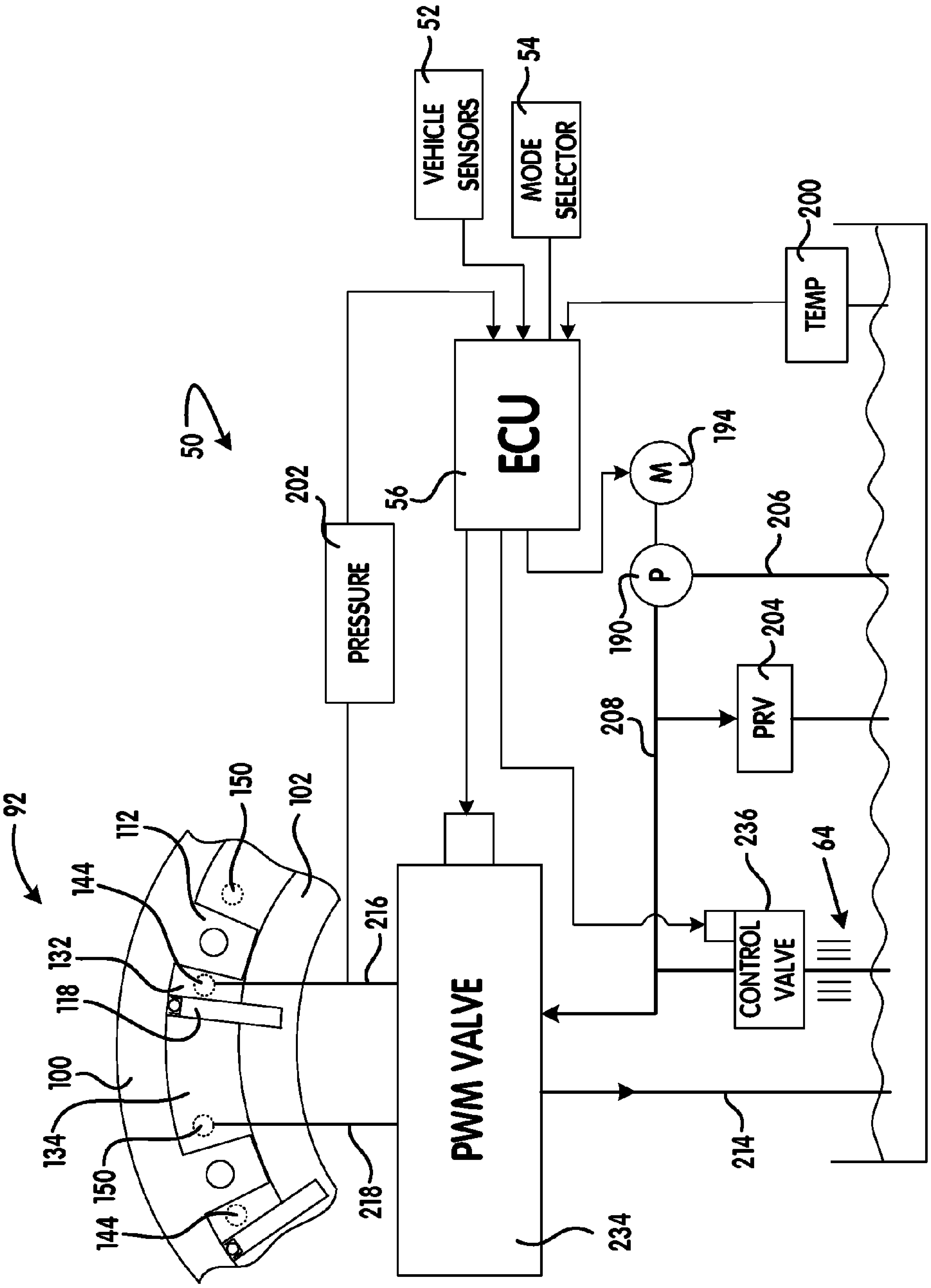

VEHICLE SENSORS
52
MODE SELECTOR
54
TEMP
200
ECU
56
50
PRESSURE
202
M
194
P
190
206
PRV
204
208
CONTROL VALVE
236
64
PWM VALVE
234
216
218
214
92
112
150
102
144
132
118
100
134
150
144

FIG. 8

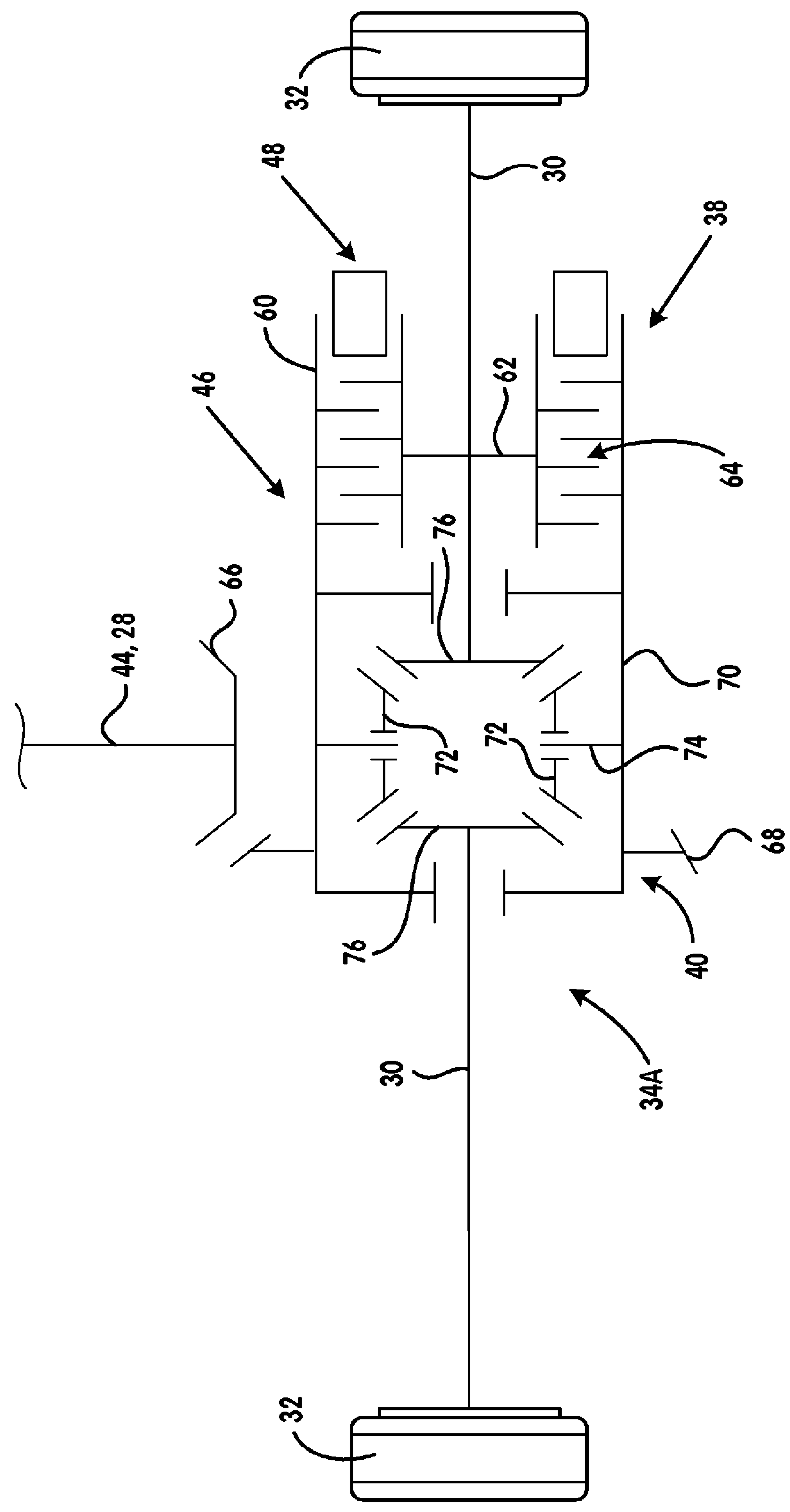


FIG. 9

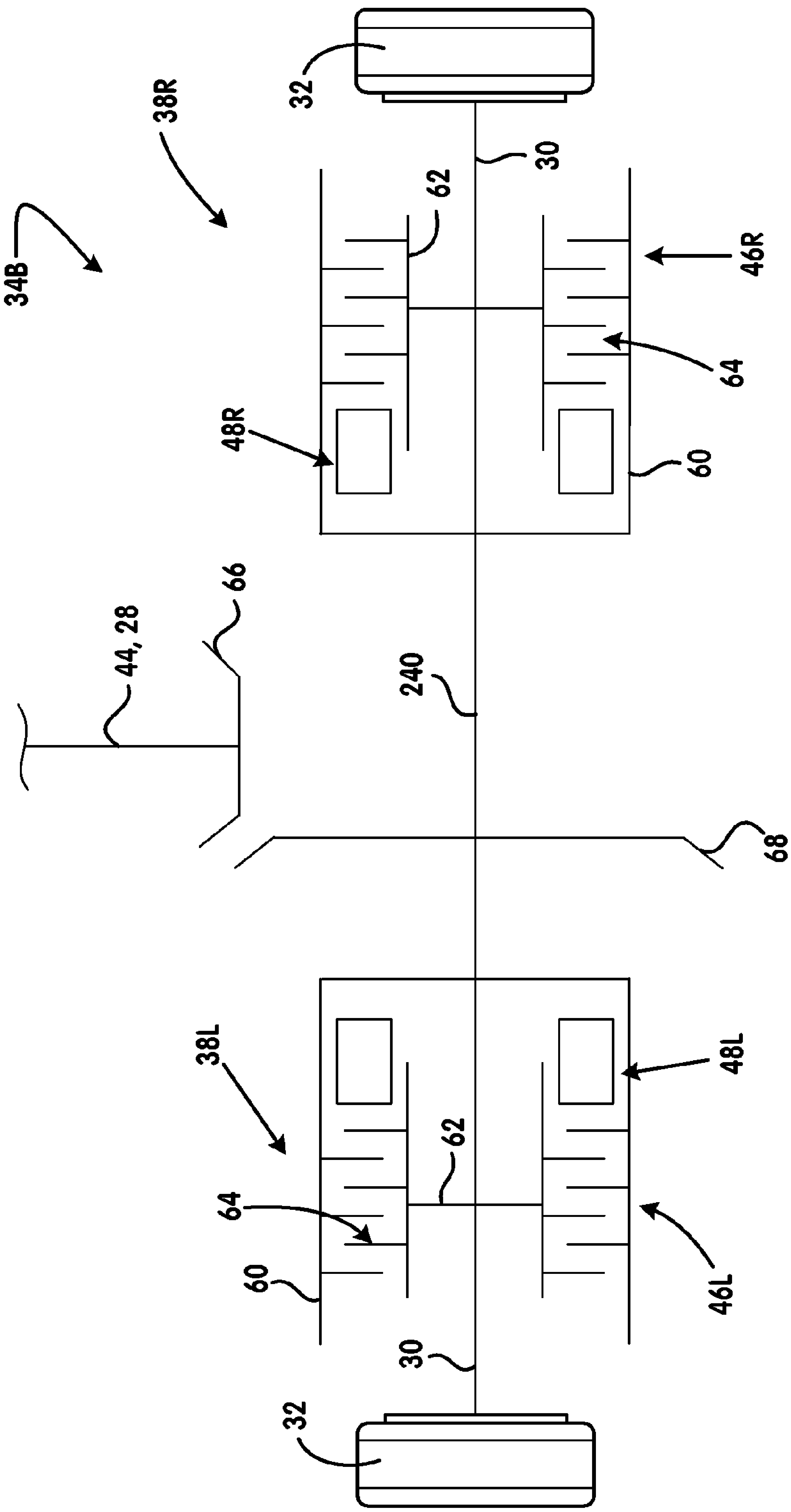

34B
38R
32
30
62
46R
64
48R
60
66
44, 28
240
68
38L
48L
62
64
60
46L
30
32

FIG. 10

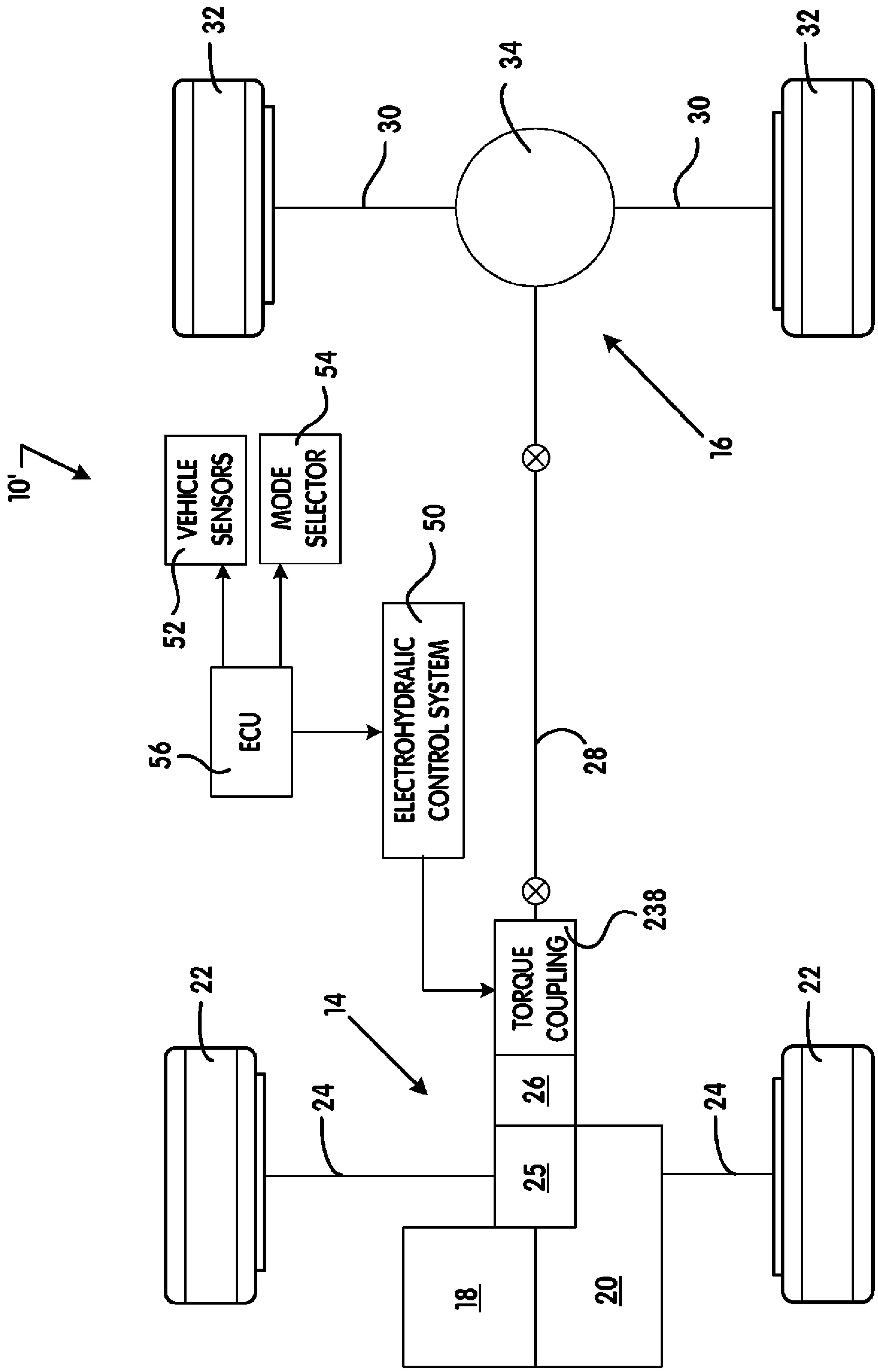
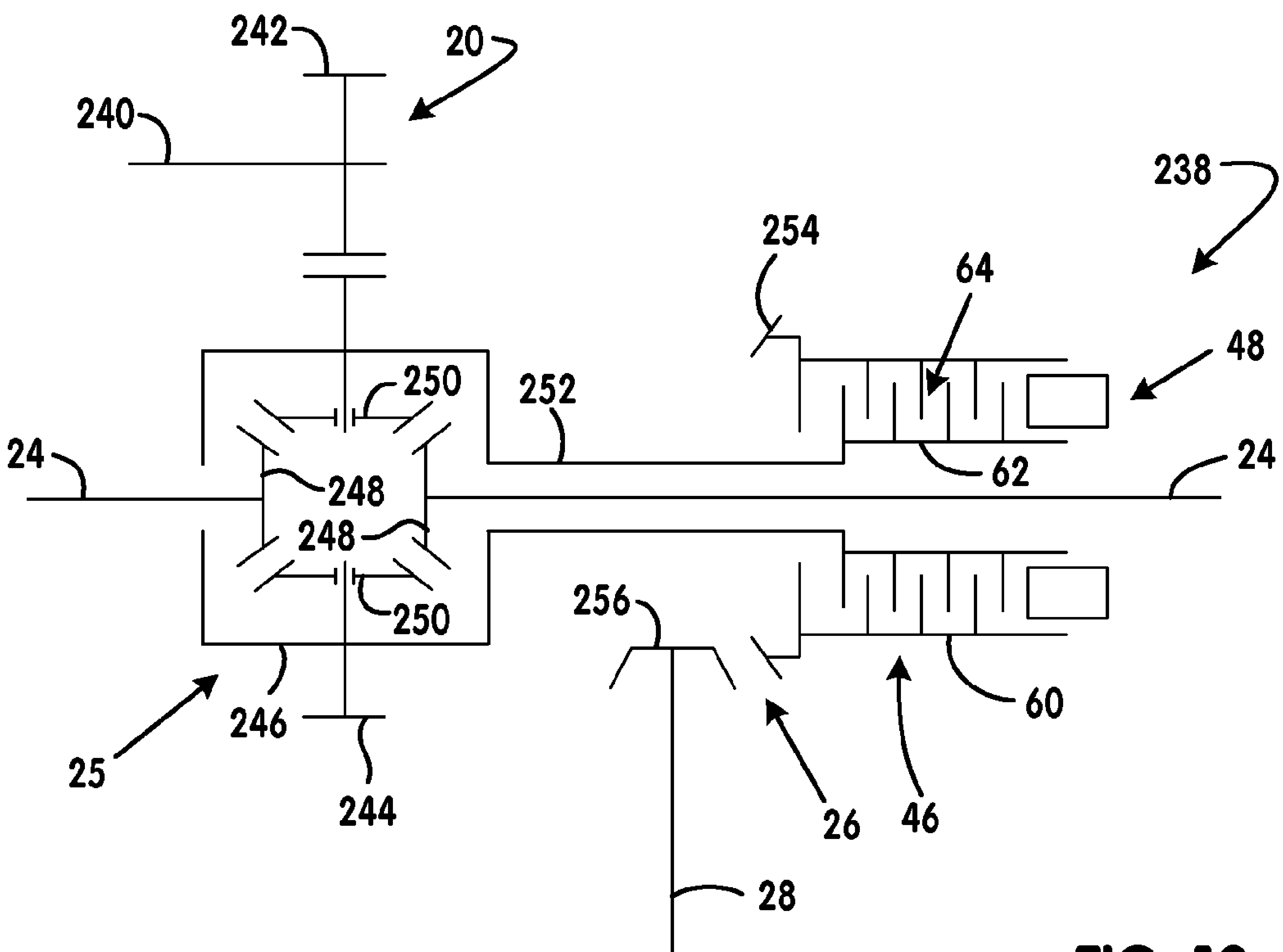

10'
32
34
30
32
30
VEHICLE SENSORS
MODE SELECTOR
54
52
50
ELECTROHYDRALIC CONTROL SYSTEM
ECU
56
16
28
238
TORQUE COUPLING
22
14
24
26
25
18
20
22
24

FIG. 11


242
20
240
238
254
64
48
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252
24
248
248
62
24
250
256
246
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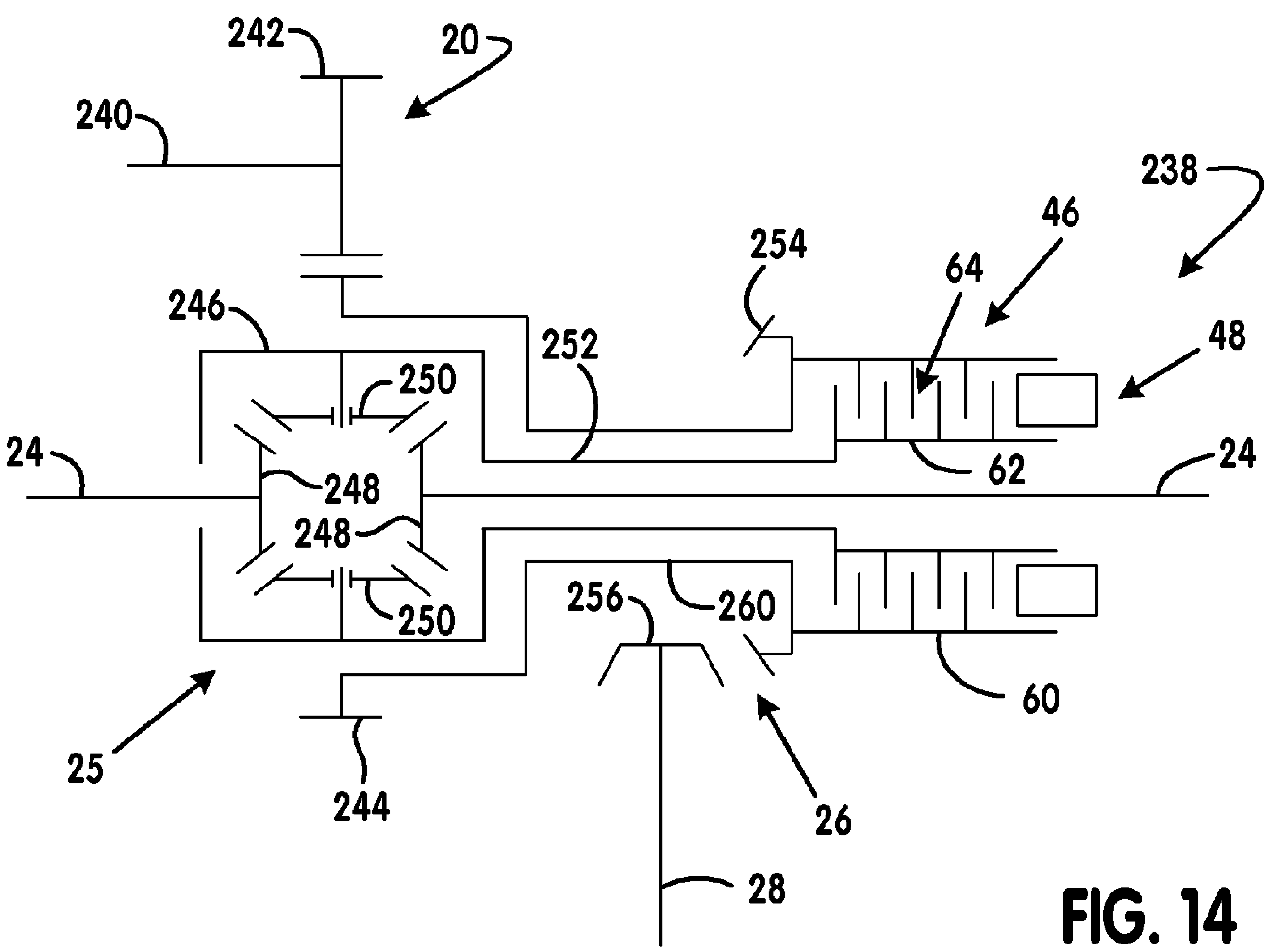
25
244
26
46
28

FIG. 12


242
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240
238
46
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24
248
62
24
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256
260
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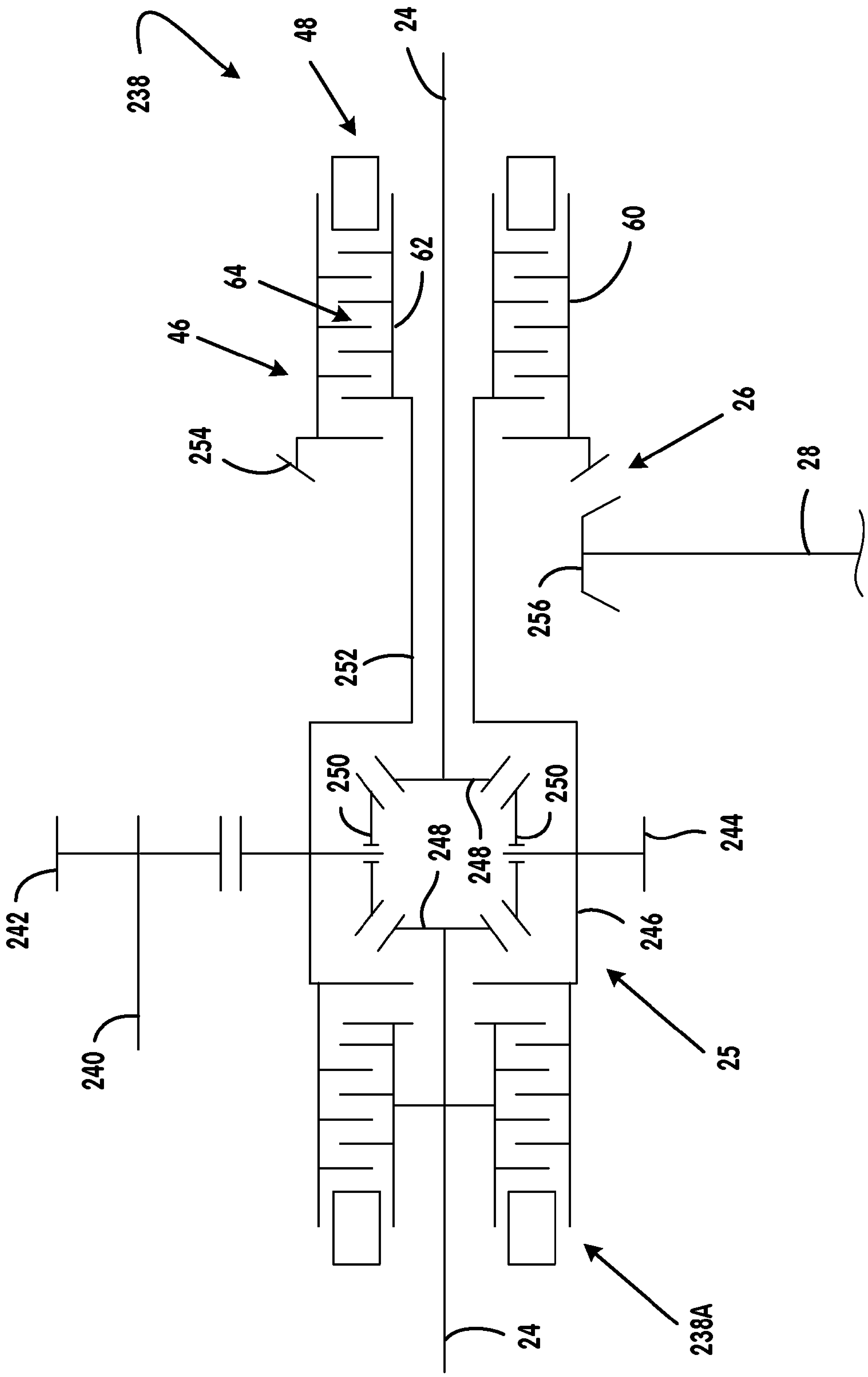
25
244
26
28

FIG. 14


238
48
24
60
62
64
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244
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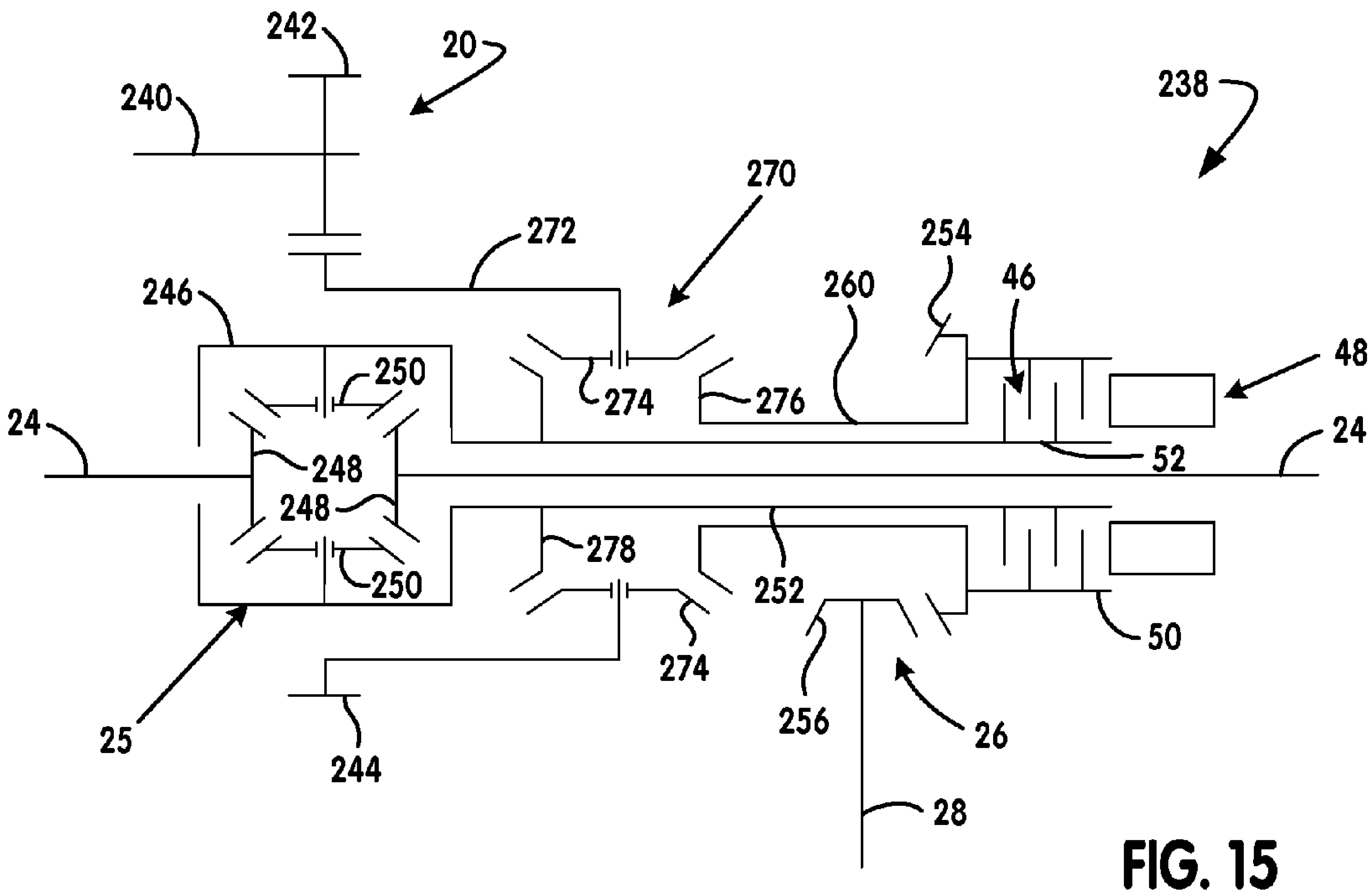
25
24
238A

FIG. 13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278
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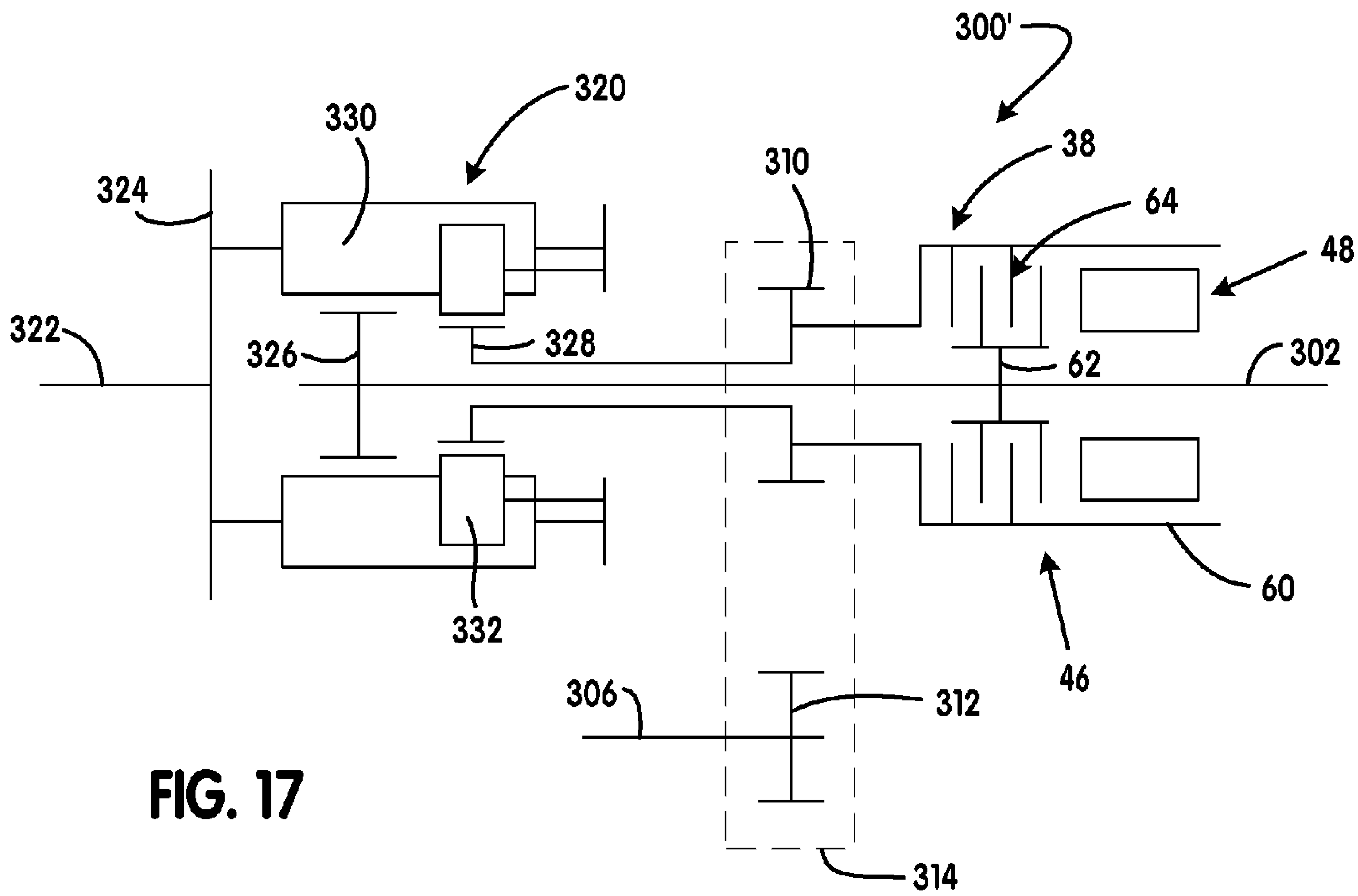
252
50
274
256
25
26
244
28

FIG. 15


300'
320
330
38
310
64
324
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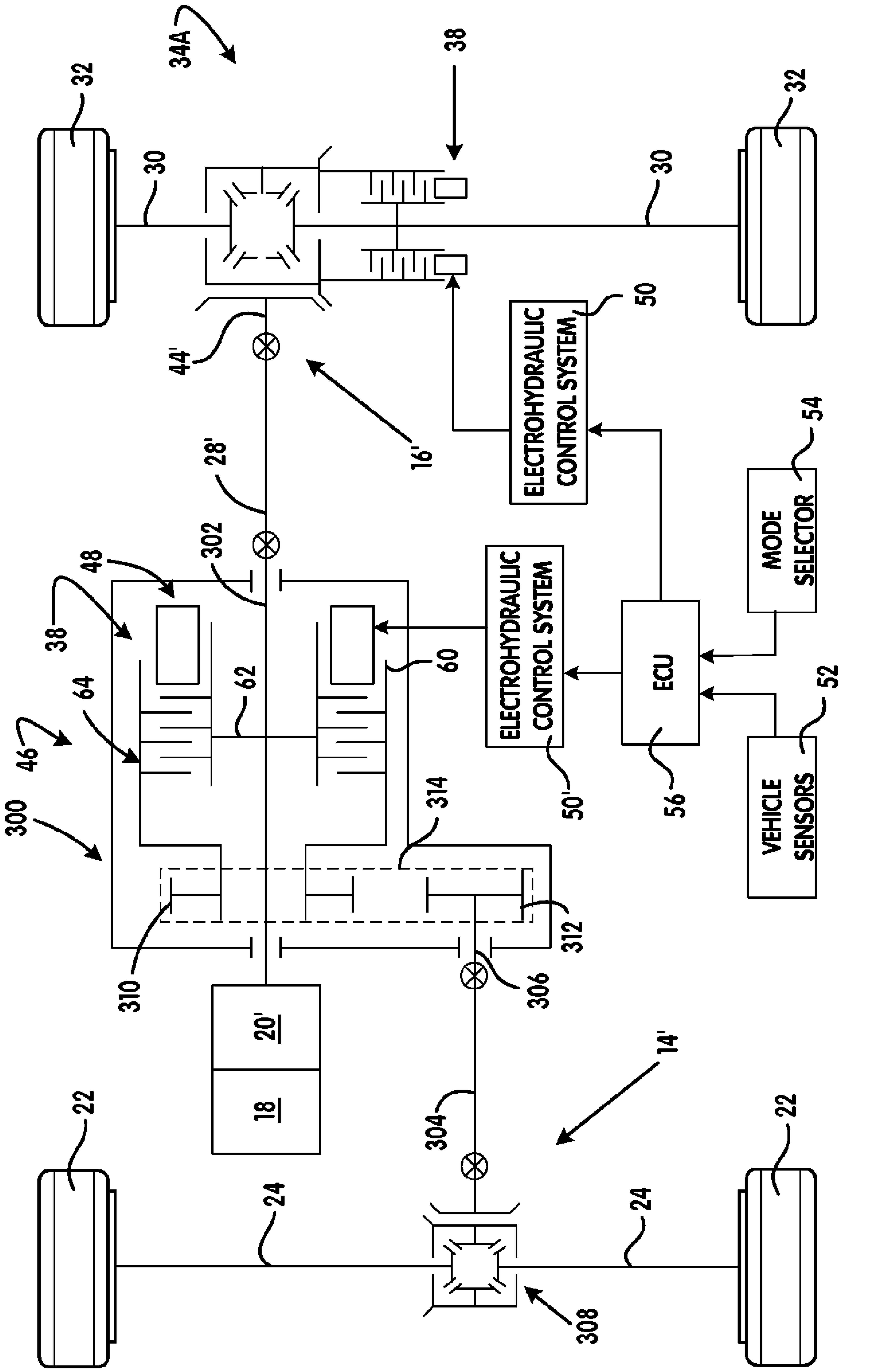
322
328
326
62
302
60
332
46
306
312
314

FIG. 17


34A
38
32
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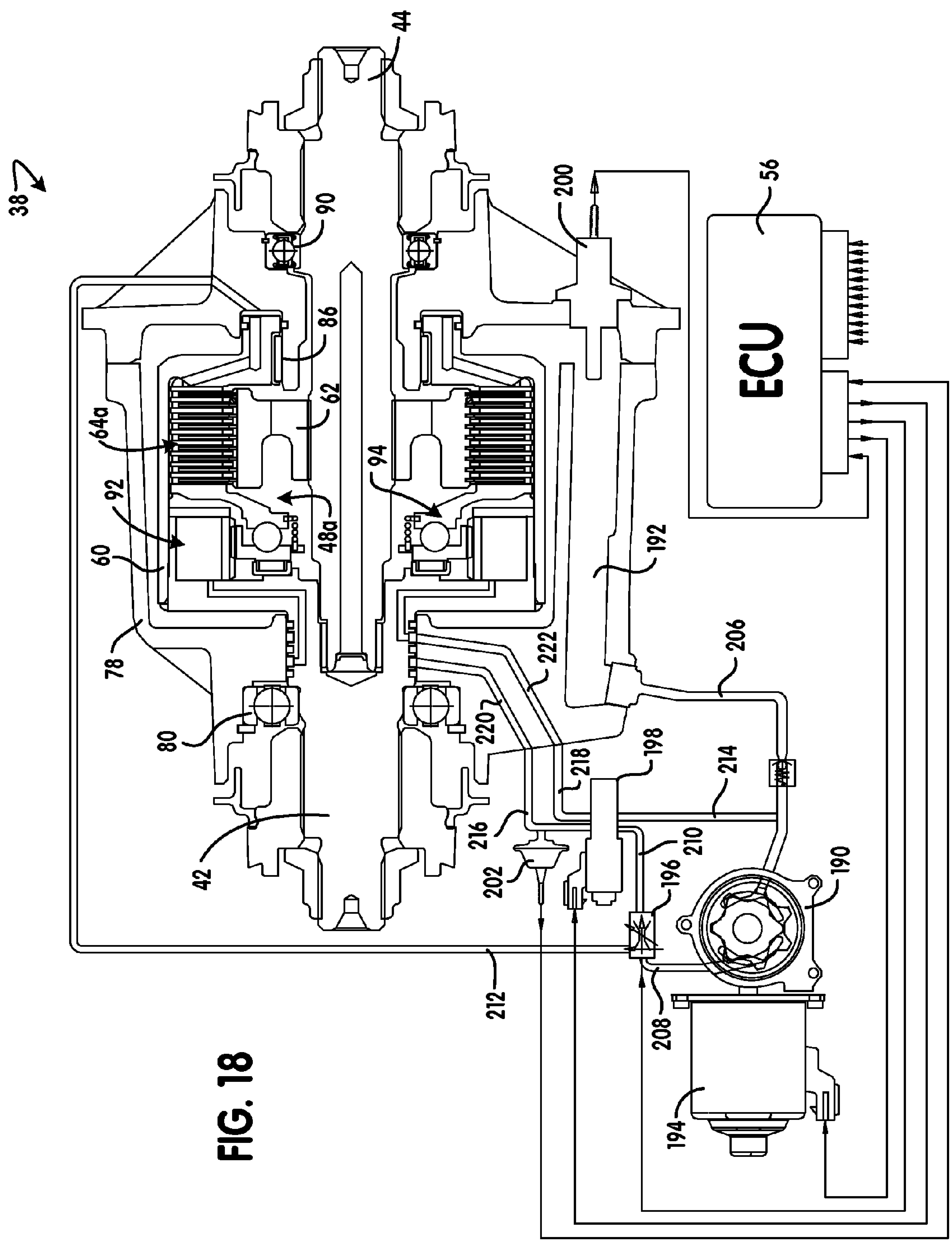
30
32
44'
50
ELECTROHYDRAULIC CONTROL SYSTEM
16'
28'
54
MODE SELECTOR
302
48
38
ELECTROHYDRAULIC CONTROL SYSTEM
ECU
60
62
64
46
52
VEHICLE SENSORS
50'
56
314
300
310
312
306
20'
18
14'
304
22
22
24
24
308

FIG. 16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ECU
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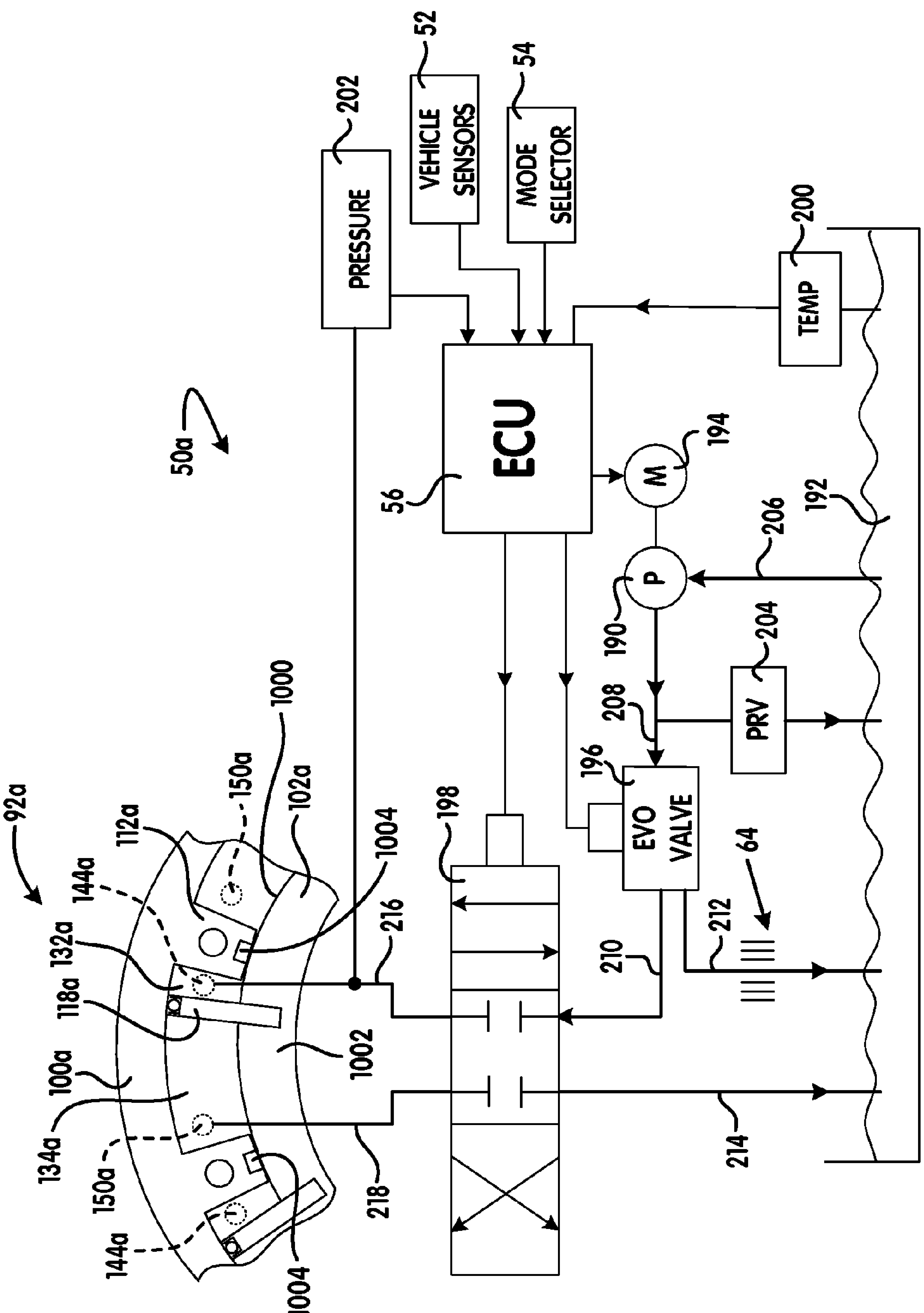
212
208
194

FIG. 18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PRESSURE
202
VEHICLE SENSORS
52
MODE SELECTOR
54
TEMP
200
ECU
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M
194
P
190
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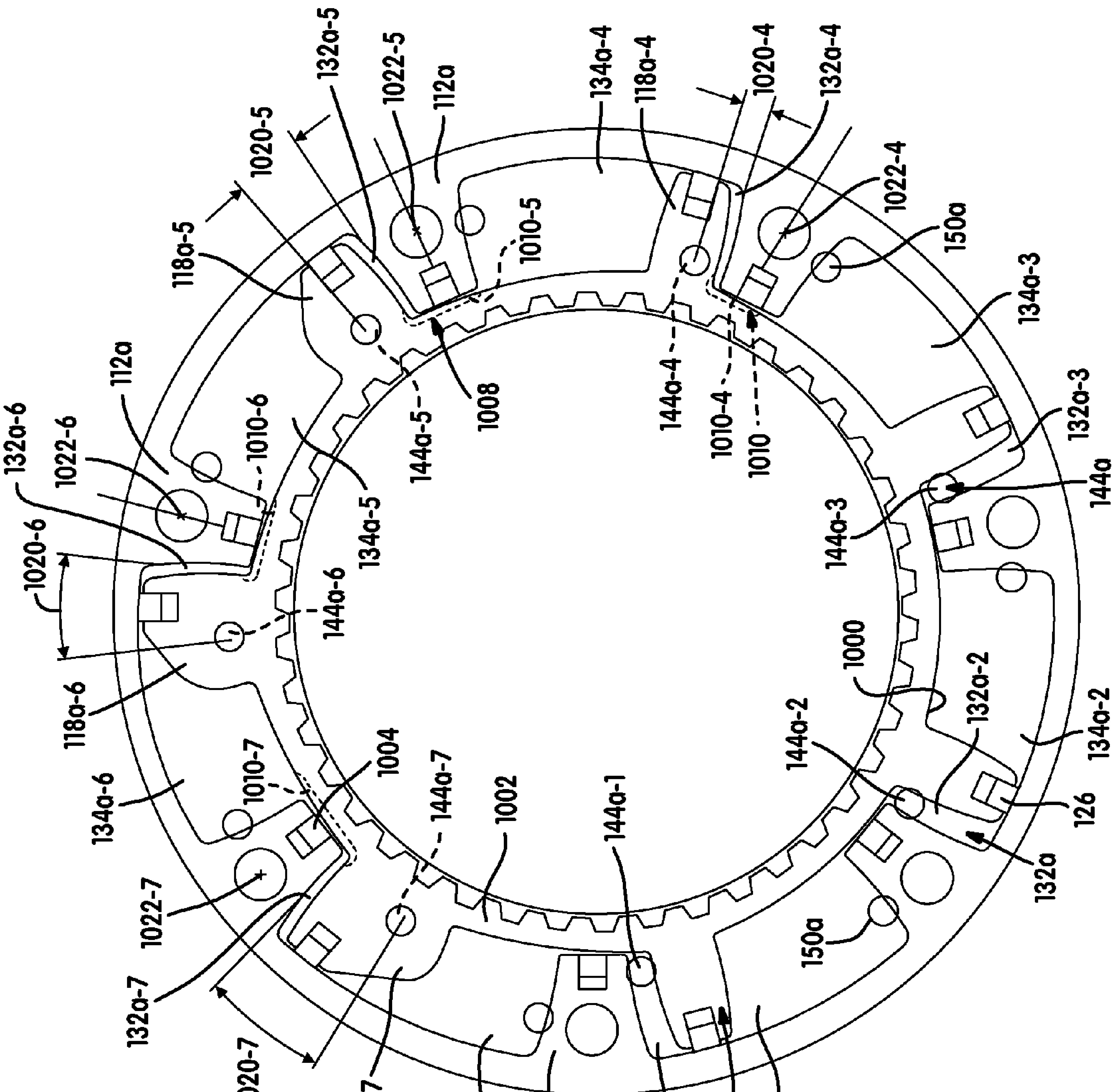
192
PRV
204
208
EVO VALVE
196
64
212
210
198
216
214
218
1000
1002
1004
92a
102a
150a
112a
144a
132a
118a
100a
134a

FIG. 19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1010-5
134a-3
1008
144a-4
1010-4
1010
132a-3
112a
144a-5
134a-5
1010-6
1022-6
132a-6
144a
144a-3
1020-6
144a-6
1000
132a-2
118a-6
134a-2
144a-2
1004
1010-7
134a-6
144a-7
1002
126
132a
144a-1
1022-7
150a
132a-7
1020-7
118a-7
134a-7
112a
132a-1
118a
134a

FIG. 20

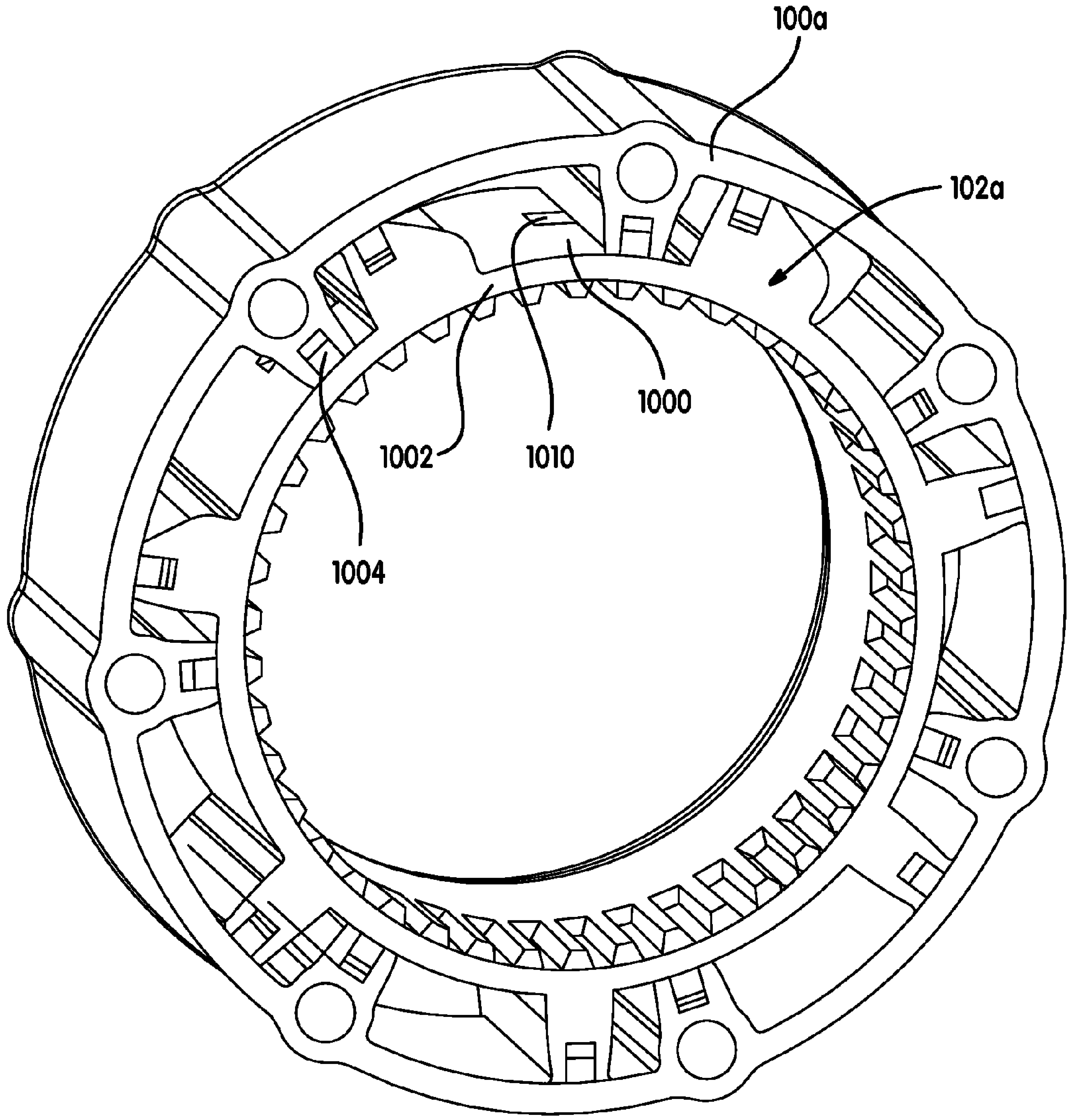

FIG. 21

ACTIVE TORQUE COUPLING WITH HYDRAULICALLY-ACTUATED ROTARY CLUTCH ACTUATOR

CROSS-REFERENCE TO RELATED APPLICATIONS

This application is a continuation-in-part of U.S. patent application Ser. No. 10/771,664 filed on Feb. 4, 2004, now U.S. Pat. No. 6,945,374.

FIELD OF THE INVENTION

The present invention relates generally to power transfer systems for controlling the distribution of drive torque between the front and rear drivelines of a four-wheel drive vehicle and/or the left and right wheels of an axle assembly. More particularly, the present invention is directed to a power transmission device for use in motor vehicle driveline applications having a torque transfer mechanism equipped with a clutch actuator that is operable for controlling actuation of a multi-plate friction clutch.

BACKGROUND OF THE INVENTION

In view of increased demand for four-wheel drive vehicles, a plethora of power transfer systems are currently being incorporated into vehicular driveline applications for transferring drive torque to the wheels. In many vehicles, a power transmission device is operably installed between the primary and secondary drivelines. Such power transmission devices are typically equipped with a torque transfer mechanism for selectively and/or automatically transferring drive torque from the primary driveline to the secondary driveline to establish a four-wheel drive mode of operation. For example, the torque transfer mechanism can include a dog-type lock-up clutch that can be selectively engaged for rigidly coupling the secondary driveline to the primary driveline to establish a "part-time" four-wheel drive mode. When the lock-up clutch is released, drive torque is only delivered to the primary driveline for establishing a two-wheel drive mode.

A modern trend in four-wheel drive motor vehicles is to equip the power transmission device with an adaptively controlled transfer clutch in place of the lock-up clutch. The transfer clutch is operable for automatically directing drive torque to the secondary wheels, without any input or action on the part of the vehicle operator, when traction is lost at the primary wheels for establishing an "on-demand" four-wheel drive mode. Typically, the transfer clutch includes a multi-plate clutch assembly that is installed between the primary and secondary drivelines and a clutch actuator for generating a clutch engagement force that is applied to the clutch assembly. The clutch actuator can be a power-operated device that is actuated in response to electric control signals sent from an electronic controller unit (ECU). The electric control signals are typically based on changes in current operating characteristics of the vehicle (i.e., vehicle speed, interaxle speed difference, acceleration, steering angle, etc.) as detected by various sensors. Thus, such "on-demand" transfer clutch can utilize adaptive control schemes for automatically controlling torque distribution during all types of driving and road conditions. Such adaptively controlled transfer clutches can also be used in association with a center differential operably installed between the primary and secondary drivelines for automatically controlling interaxle slip and torque biasing in a full-time four-wheel drive application.

A large number of adaptively controlled transfer clutches have been developed with an electro-mechanical clutch actuator that can regulate the amount of drive torque transferred to the secondary driveline as a function of the electric control signal applied thereto. In some applications, the transfer clutch employs an electromagnetic clutch as the power-operated clutch actuator. For example, U.S. Pat. No. 5,407,024 discloses a electromagnetic coil that is incrementally activated to control movement of a ball-ramp drive assembly for applying a clutch engagement force to the multi-plate clutch assembly. Likewise, Japanese Laid-open Patent Application No. 62-18117 discloses a transfer clutch equipped with an electromagnetic clutch actuator for directly controlling actuation of the multi-plate clutch pack assembly. As an alternative, the transfer clutch can employ an electric motor and a mechanical drive assembly as the power-operated clutch actuator. For example, U.S. Pat. No. 5,323,871 discloses a transfer clutch equipped with an electric motor that controls rotation of a sector plate which, in turn, controls pivotal movement of a lever arm that is operable for applying the clutch engagement force to the multi-plate clutch assembly. Likewise, Japanese Laid-open Patent Application No. 63-66927 discloses a transfer clutch which uses an electric motor to rotate one cam plate of a ball-ramp operator for engaging the multi-plate clutch assembly. Finally, U.S. Pat. Nos. 4,895,236 and 5,423,235 respectively disclose a transfer clutch having an electric motor which drives a reduction gearset for controlling movement of a ball screw operator and a ball-ramp operator which, in turn, apply the clutch engagement force to the clutch assembly.

In contrast to the electro-mechanical clutch actuators discussed previously, it is also well known to equip the transfer clutch with an electro-hydraulic clutch actuator. For example, U.S. Pat. Nos. 4,862,769 and 5,224,906 generally disclose use of an electric motor or solenoid to control the fluid pressure exerted by an apply piston on a multi-plate clutch assembly. In addition, U.S. Pat. No. 6,520,880 discloses a hydraulic actuation system for controlling the fluid pressure supplied to a hydraulic motor arranged which is associated with a differential gear mechanism in a drive axle assembly.

While many adaptive clutch actuation systems similar to those described above are currently used in four-wheel drive vehicles, a need exists to advance the technology and address recognized system limitations. For example, the size and weight of the friction clutch components and the electrical power requirements of the clutch actuator needed to provide the large clutch engagement loads make many systems cost prohibitive for use in most four-wheel drive vehicle applications. In an effort to address these concerns, new technologies are being developed for use in power-operated clutch actuator applications.

SUMMARY OF THE INVENTION

Thus, its is an objective of the present invention to provide a power transmission device for use in a motor vehicle having a torque transfer mechanism equipped with a unique electrohydraulically-operated clutch actuator that is operable to control adaptive engagement of a multi-plate clutch assembly.

As a related objective of the present invention, the torque transfer mechanism is well-suited for use in motor vehicle driveline applications to adaptively control the transfer of drive torque between first and second rotary members.

According to each preferred embodiment of the present invention, a torque transfer mechanism and an electrohydraulic control system are disclosed for adaptively controlling the transfer of drive torque between first and second rotary members in a power transmission device of the type used in motor vehicle driveline applications. The torque transfer mechanism includes a rotary input member that is configured to receive drive torque from a source of drive torque, a rotary output member adapted to transmit drive torque to an output device and a torque transmission mechanism that is configured to transfer drive torque from said input member to said output member. The torque transmission mechanism can include a friction clutch that can be operably disposed between the input member and the output member and an electrohydraulic control system for controlling engagement of the friction clutch. The electrohydraulic control system can include a clutch actuator, a fluid pump, a motor driving the fluid pump, and a controller. The clutch actuator can include a rotary operator and a thrust mechanism. The rotary operator can have first and second components with a plurality of actuation chambers and a plurality of return chambers. The first component can be fixed for rotation with one of the input and output members and the second component can be configured to rotate relative to the first component. The thrust mechanism can be operable for applying a clutch engagement force on the friction clutch in response to rotation of the second component relative to the first component. The controller can be operable for controlling the fluid pressure supplied to at least one of the actuation chambers and the return chambers to thereby control rotation of the second component relative to the first component of the rotary operator. The electrohydraulic control system is configured so that the actuation chambers are divided into at least two groups. A first group is configured such that the activation chamber or activation chambers included therein contain a relatively high pressure fluid when the second component is rotated relative to the first component in an actuating direction regardless of a relative position of the first and second components. A second group is configured such that the activation chamber or activation chambers included therein contain a relatively low pressure fluid when the second component is positioned relative to the first component within a predetermined actuating interval and contain the relatively high pressure fluid when the second component is rotated relative to the first component in the actuation direction by an amount that exceeds the predetermined actuating interval.

BRIEF DESCRIPTION OF THE DRAWINGS

Further objectives, features and advantages of the present invention will become apparent to those skilled in the art from analysis of the following written description, the appended claims, and the accompanying drawings in which:

FIG. **1** illustrates the drivetrain of a four-wheel drive vehicle equipped with a power transmission device according to the present invention;

FIG. **2** is a schematic illustration of the power transmission device equipped with a torque transfer mechanism embodying the inventive concepts of the present invention;

FIGS. **3** and **3**A are sectional views of the torque transfer mechanism constructed in accordance with a preferred embodiment of the present invention;

FIG. **4** is a sectional view of the rotary operator associated with the torque transfer mechanism shown in FIGS. **3** and **3**A;

FIG. **5** is a schematic diagram of the electrohydraulic control circuit associated with the torque transfer mechanism of the present invention;

FIGS. **6** through **8** are schematic diagrams of alternative electrohydraulic control systems adapted for use with the torque transfer mechanism of the present invention;

FIG. **9** is a schematic illustration of an alternative power transmission device available for use with the drivetrain shown in FIG. **1**;

FIG. **10** is a schematic illustration of another alternative embodiment of a power transmission device according to the present invention;

FIG. **11** illustrates an alternative drivetrain arrangement for a four-wheel drive motor vehicle equipped with another power transmission device embodying the present invention;

FIGS. **12** through **15** schematically illustrate different embodiments of the power transmission device shown in FIG. **11**;

FIG. **16** is an illustration of another drivetrain arrangement for a four-wheel drive vehicle equipped with a power transmission device embodying the present invention;

FIG. **17** is a schematic illustration of an alternative construction for the power transmission device shown in FIG. **16**;

FIG. **18** is a sectional view of the torque transfer mechanism constructed in accordance with the teachings of the present invention;

FIG. **19** is a schematic illustration of the power transmission device equipped with the torque transfer mechanism of FIG. **18**;

FIG. **20** is a perspective view of a portion of the torque transfer mechanism of FIG. **18** illustrating the clutch actuator in greater detail; and

FIG. **21** is a sectional view of a portion of the torque transfer mechanism of FIG. **18** illustrating the construction of the reaction ring and the actuator ring.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

The present invention is directed to a torque transfer mechanism that can be adaptively controlled for modulating the torque transferred from a first rotary member to a second rotary member. The torque transfer mechanism finds particular application in power transmission devices for use in motor vehicle drivelines such as, for example, a torque transfer clutch in a transfer case, a power take-off unit or an in-line torque coupling, a torque biasing clutch associated with a differential unit in full-time transfer cases or power take-off unit or in a drive axle assembly, or any other possible torque transfer application. Thus, while the present invention is hereinafter described in association with particular power transmission devices for use in specific driveline applications, it will be understood that the arrangements shown and described are merely intended to illustrate embodiments of the present invention.

With reference to FIG. **1**, a schematic layout of a vehicular drivetrain **10** is shown to include a powertrain **12**, a first or primary driveline **14** driven by powertrain **12**, and a second or secondary driveline **16**. Powertrain **12** includes an engine **18** and a multi-speed transaxle **20** arranged to normally provide motive power (i.e., drive torque) to a pair of first wheels **22** associated with primary driveline **14**. Primary driveline **14** further includes a pair of axleshafts **24** connecting wheels **22** to a differential unit **25** associated with transaxle **20**. Secondary driveline **16** includes a power take-off unit (PTU) **26** driven by the output of transaxle **20**, a propshaft **28** driven by PTU **26**, a pair of axleshafts **30** connected to a pair of second wheels **32**, and a drive axle assembly **34** that is operable to selectively transfer drive torque from propshaft **28** to axle halfshafts **30**.

Drive axle assembly **34** is a power transmission device according to one preferred embodiment of the present invention. In particular, drive axle assembly **34** is shown schematically in FIG. **2** to include a housing assembly **36** which encloses a torque transfer mechanism **38** and a differential unit **40**. Torque transfer mechanism **38** functions to selectively transfer drive torque from propshaft **28** to an input component of differential unit **40**. Specifically, the torque transfer mechanism, hereinafter referred to as torque coupling **38**, includes an input shaft **42** driven by propshaft **28**, a pinion shaft **44**, a transfer clutch **46** operably connected between input shaft **42** and pinion shaft **44**, and a clutch actuator **48** for engaging transfer clutch **46**.

With continued reference to the drawings, drivetrain **10** is shown to further include an electronically-controlled power transfer system for permitting a vehicle operator to select between a two-wheel drive mode, a locked ("part-time") four-wheel drive mode, and an adaptive ("on-demand") four-wheel drive mode. In this regard, transfer clutch **46** can be selectively engaged for transferring drive torque from input shaft **42** to pinion shaft **44** for establishing both of the part-time and on-demand four-wheel drive modes. The power transfer system includes an electrohydraulic control system **50** for selectively actuating clutch actuator **48**, vehicle sensors **52** for detecting certain dynamic and operational characteristics of the motor vehicle, a mode select mechanism **54** for permitting the vehicle operator to select one of the available drive modes, and an electronic control unit (ECU) **56** for controlling operation of the components associated with electrohydraulic control system **50** which, in turn, controls actuation of clutch actuator **48** in response to input signals from vehicle sensors **52** and mode selector **54**.

Referring primarily to FIGS. **2** and **3**, transfer clutch **46** generally includes a first clutch member **60** driven by input shaft **42**, a second clutch member **62** driving pinion shaft **44**, and a multi-plate clutch pack **64** of alternately interleaved clutch plates installed between input shaft **42** and pinion shaft **44**. As shown in this particular arrangement, first clutch member **60** is a clutch drum fixed for rotation with input shaft **42** and second clutch member **62** is a clutch hub fixed (i.e., splined) for rotation with pinion shaft **44**. Pinion shaft **44** drives a pinion gear **66** meshed with a ring gear **68** which, in turn, drives differential unit **40**. Differential unit **40** includes a carrier **70** driven by ring gear **68**, a pair of pinion gears **72** rotatably supported on pinion posts **74** fixed to carrier **70**, and a pair of side gears **76**. Side gears **76** are meshed with both pinion gears **74** and are coupled for rotation with a corresponding one of axleshafts **30**.

Referring now to FIGS. **3** and **3A**, a preferred construction for torque coupling **38** will now be described in greater detail. Torque coupling **38** includes a case assembly **78** that is mounted in or forms part of housing **36** of drive axle assembly **34**. A bearing assembly **80** supports input shaft **42** for rotation relative to case assembly **78**. In addition, input shaft **42** is shown to include an integral end plate **82** that is rigidly secured (i.e., welded) to clutch drum **60**. An end plate segment **84** of drum **60** is rotatively supported by a bearing assembly **86** from case assembly **78**. Pinion shaft **44** has a first end rotatably supported by a bushing or bearing assembly **88** in a central bore formed in input shaft **42** while its second end extends out of end plate segment **84** of drum **60** and is rotatably supported from case assembly **78** by a bearing assembly **90**. Clutch actuator **48** generally includes a rotary operator **92**, a thrust mechanism **94**, and an apply plate **96**. Apply plate **96** is secured (i.e., splined) for rotation with drum **60** of transfer clutch **46**.

As will be detailed, clutch actuator **48** is operable for generating and exerting a compressive clutch engagement force on clutch pack **64**. Such engagement of clutch pack **64** causes rotary power ("drive torque") to be transferred from input shaft **42** to pinion shaft **44**. Specifically, clutch actuator **48** is operable for controlling axial movement of apply plate **96** and thus, the magnitude of the clutch engagement force applied to clutch pack **64**. In particular, apply plate **96** is axially moveable relative to clutch pack **64** between a first or "released" position and a second or "locked" position. With apply plate **96** in its released position, a minimum clutch engagement force is exerted on clutch pack **64** such that virtually no drive torque is transferred from input shaft **42** through transfer clutch **46** to pinion shaft **44**, thereby establishing the two-wheel drive mode. In contrast, movement of apply plate **96** to its locked position causes a maximum clutch engagement force to be applied to clutch pack **64** such that pinion shaft **44** is, in effect, coupled for common rotation with input shaft **42**, thereby establishing part-time four-wheel drive mode. Accordingly, controlling the position of apply plate **96** between its released and locked positions permits adaptive regulation of the amount of drive torque transferred from input shaft **42** to pinion shaft **44**, thereby establishing the on-demand four-wheel drive mode.

Rotary operator **92** includes a first or reaction ring **100** that is concentrically aligned with a second or "actuator" ring **102**. The rings are retained in an annular chamber for **104** defined between end plate **82** and a retainer plate **106**. While not shown, retainer plate **106** is secured by a plurality of bolts to end plate **82** which also extend through mounting holes **108** in reaction ring **100**. As such, reaction ring **100** is fixed for common rotation with input shaft **42**.

As best seen from FIG. **4**, reaction ring **100** includes a cylindrical body segment **110** and a plurality of radially inwardly projecting lugs **112** which define a complimentary number of longitudinally extending channels **114** therebetween. Likewise, actuator ring **102** has a cylindrical body segment **116** and a plurality of separator plates **118** that are retained in longitudinal slots **120** formed in body segment **116**. An outer edge surface **122** of each separator plate **118** is aligned to be in sliding engagement with an inner wall surface **124** of channels **114**. A seal strip **126** may be attached to edge surface **122** to provide a sliding fluid-tight interface between separator plates **118** and body segment **110** of reaction ring **100**. Likewise, a terminal edge surface **128** of each lug **112** is aligned to be in sliding engagement with an outer wall surface **130** of actuator ring **102**. Each separator plate **118** defines an actuation chamber **132** and a return chamber **134** between adjacent pairs of lugs **112**. As such, a plurality of circumferentially-spaced alternating actuation chambers **132** and return chambers **134** are established in association with rotary operator **92**.

To provide means for supplying hydraulic fluid from electrohydraulic control system **50** to actuation chambers **132**, a first flow path is formed in input shaft **42** and its end plate segment **82**. The first flow passage includes an annular chamber **140** which communicates with a series of circumferentially-spaced flow passages **142** having ports **144** in fluid communication with actuation chambers **132**. Similarly, a second flow path provides means for supplying hydraulic fluid from control system **50** to return chambers **134**. This second flow path includes an annular chamber **146** which communicates with a series of circumferentially-spaced flow passages **148** having ports **150** in fluid communication with return chambers **134**. As will be detailed, increasing the fluid pressure delivered through ports **144** to actuation chambers **132** while decreasing the fluid pressure delivered through ports **150** to return chambers **134** causes actuator ring **102** to move (i.e., index) in a first rotary direction (i.e., counterclockwise) relative to reaction ring **100** for causing thrust mechanism **94** to urge apply plate **96** to move toward its locked position. In contrast, decreasing the fluid pressure in actuation chambers **132** and increasing the fluid pressure in return chambers **134** causes actuator ring **102** to index in a second rotary direction (i.e., clockwise) relative to reaction ring **100** for causing thrust mechanism **94** to permit apply plate **96** to move toward its released position.

With continued reference primarily to FIG. **3**A, thrust mechanism **94** is shown to be a ball ramp unit having a first cam member **152**, a second cam member **154**, and rollers, such as balls **156**. First cam member **152** is fixed via a spline connection **158** for rotary movement with actuator ring **102**. Likewise, second cam member **154** is fixed to apply plate **96** for common rotation with drum **60** and input shaft **42**. Each ball **156** is disposed in a cam channel defined between a cam track **170** formed in first cam member **152** and a corresponding cam track **172** formed in second cam member **154**. Preferably, a plurality of cam channels are provided between first cam member **152** and second cam member **154** with cam tracks **170** and **172** configured as oblique sections of a helical torus. However, balls **156** and cam tracks **170** and **172** may be replaced with alternative cam components and/or ramp configurations that function to cause axial displacement of second cam member **154**. In any arrangement, the load transferring components can not be self-locking or self-engaging so as to permit fine control of the translational movement of apply plate **96** (via second cam member **154**) for precise control of the engagement characteristics (i.e., torque transfer) of transfer clutch **46**. As seen, a thrust bearing **176** is located between end plate **82** and first cam member **152**.

Ball ramp unit **94** further includes a torsional return spring **178** that is operably disposed between cam members **152** and **154**. Return spring **178** functions to angularly bias cam members **152** and **154** to return to a "retracted" position for de-energizing ball ramp unit **94**. Such angular movement of the cam members to the retracted position due to the biasing of return spring **178** results in angular movement of actuator ring **102** relative to reaction ring **102** in the second angular direction toward a first or "low pressure" position and translational movement of apply plate **96** toward its released position. With actuator ring **102** in its low pressure position (as shown in FIG. **4**), ball ramp unit **94** is de-energized and apply plate **96** is in its released position so as to exert a predetermined minimum clutch engagement force on clutch pack **64** for releasing engagement of transfer clutch **46**.

Electrohydraulic control system **50** is operable to supply high pressure fluid to actuation chambers **132** for causing actuator ring **102** to rotate relative to reaction ring **100** in the first direction from its low pressure position toward a second or "high pressure" position. Such movement of actuator ring **102** results in corresponding relative angular movement between cam members **152** and **154** from the retracted position toward a second or "extended" position for energizing ball ramp unit **94**. Accordingly, the profile of cam tracks **170** and **172** establishes the resultant amount of translational movement of second cam member **154** required to cause corresponding axial movement of apply plate **96** from its released position toward its locked position. When actuator ring **102** is in its high pressure position, ball ramp unit **94** is fully energized and apply plate **96** is located in its locked position such that the maximum clutch engagement force is exerted on clutch pack **64** for fully engaging transfer clutch **46**. Furthermore, electrohydraulic control system **50** is operable to supply high pressure fluid to return chambers **134** and vent actuation chambers **132** for causing actuator ring **102** to rotate relative to reaction ring **100** in the second direction from its high pressure position. Such angular movement of actuator ring **102** results in corresponding relative angular movement between cam members **152** and **154** from the extended position toward the retracted position for releasing ball ramp unit **94**. As such, apply plate **96** is caused to move from its locked position toward its released position for releasing engagement of transfer clutch **46**.

With apply plate **96** in its released position, virtually no drive torque is transferred from input shaft **42** to pinion shaft **44** through torque coupling **38** so as to effectively establish the two-wheel drive mode. In contrast, location of apply plate **96** in its locked position results in a maximum amount of drive torque being transferred to pinion shaft **44** for coupling pinion shaft **44** for common rotation with input shaft **42**, thereby establishing the part-time four-wheel drive mode. Accordingly, controlling the position of apply plate **96** between its released and locked positions permits variable control of the amount of drive torque transferred from input shaft **42** to pinion shaft **44** for establishing an on-demand four-wheel drive mode. Thus, the magnitude of the fluid pressure supplied to actuation chambers **132** and return chambers **134** controls the angular position of actuator ring **102** relative to reaction ring **100**, thereby controlling actuation of ball ramp unit **94** and the resulting movement of apply plate **96** between its released and locked positions.

Referring to FIGS. **3** and **5**, the components associated with electrohydraulic control system **50** will now be described. The basic components of control system **50** include a fluid pump **190** operable to draw fluid from a fluid source or sump **192** within casing **78**, an electric motor **194** driving pump **190**, an electronic variable orifice (EVO) valve **196**, and a directional control valve **198**. Control system **50** further includes a temperature sensor **200**, a pressure sensor **202** and a spring-type pressure relief valve (PRV) **204**. Motor **194** is a low power unit operable for causing pump **190** to draw fluid from sump **192** through a first flow path **206**. The magnitude of the line pressure discharged from pump **190** is delivered to an inlet of EVO valve **196** through a second flow path **208**. However, PRV **204** functions to limit the line pressure delivered to the inlet of EVO valve **196** to a predefined maximum fluid pressure value. EVO valve **196** includes a moveable valve element that functions to vary the line pressure to a "control" pressure value that is supplied from an outlet of EVO valve **196** to an inlet of control valve **198** via a third flow path **210**. As seen, the low pressure fluid by-passed through EVO valve **196** flows through a fourth flow path **212** and is used to cool and lubricate clutch pack **64**. Fluid discharged from control valve **198** is returned via a fifth flow path **214** to sump **192**. Finally, control valve **198** is shown to be in fluid communication with actuation chambers **132** via a sixth flow path **216** and in fluid communication with return chambers **134** via a seventh flow path **218**. It will be appreciated that the components of the electrohydraulic control system can be integrated into case assembly **78** to define a stand-alone assembly or, in the alternative, be remotely located and connected via suitable hoses and tubing.

Rotary operator **92** is partially shown in FIG. **5** to illustrate its hydraulic connections with control valve **198**. In the arrangement shown, EVO valve **196** is preferably an electromagnetic flow control valve that is capable of variably regulating the fluid pressure delivered to control valve **198**. Likewise, control valve **198** is shown as a 3-position shuttle valve. As seen, ECU **56** is operable to send electrical control signals to electric motor **194**, EVO valve **196** and control valve **198**. EVO valve **196** is selectively actuated by ECU **56** to control the fluid pressure supplied to third flow path **210** while control valve **198** is selectively actuated to control delivery of the pressurized fluid to either of actuation chambers **132** or return chambers **134**. As best seen from FIGS. **3** and 3A, sixth flow path **216** is in fluid communication with annular chamber **140** via a flow passage **220** formed in case **78** while seventh flow path **218** communicates with annular chamber **146** via another flow passage **222**. A plurality of ring seals **224** are provided between case **78** and input shaft **42** to delimit chambers **140** and **146**.

As contemplated by the present invention, ECU **56** is programmed to accurately control the angular position of actuator ring **102** relative to reaction ring **100** based on a predefined control strategy for transferring the desired amount of drive torque across transfer clutch **46**. The control strategy functions to determine and generate the electric control signals sent to EVO valve **196** and control valve **198** based on the mode signal from mode selector **54** and the sensor signals from sensors **52**. In addition, pressure sensor **202** sends a signal to ECU **56** that is indicative of the fluid pressure delivered through sixth flow path **216** to actuation chambers **132**. Likewise, temperature sensor **200** sends a signal to ECU **56** that is indicative of the fluid temperature in sump **192**.

In operation, if the vehicle operator selects the two-wheel drive mode, EVO valve **196** is opened and control valve **198** is initially actuated to cause third flow path **210** to be placed in communication with seventh flow path **218** while sixth flow path **216** is placed in communication with fifth flow path **214**, whereby actuation chambers **132** are vented to sump **192** while fluid at control pressure is supplied to return chambers **134**. This results in actuator ring **102** being forced to its low pressure position relative to reaction ring **100** for releasing engagement of transfer clutch **46**. Thereafter, control valve **198** can be shifted into its closed position with no fluid delivered to actuation chambers **132** or return chambers **134** since return spring **178** forcibly biases actuator ring **102** to remain in its low pressure position. In contrast, upon selection of the part-time four-wheel drive mode, EVO valve **196** is opened and control valve **198** is actuated to connect third flow path **210** to sixth flow path **216** and also connect seventh flow path **218** to fifth flow path **214**, whereby actuation chambers **132** are supplied with fluid at control pressure and return chambers **134** are vented to sump **192**. The high pressure fluid supplied to actuation chambers **132** causes actuator ring **102** to move to its high pressure position relative to reaction ring **100**, whereby ball ramp unit **94** is fully energized for moving apply plate **96** to its locked position for fully engaging transfer clutch **46**. PRV **204** functions to limit the maximum fluid pressure that can be delivered to actuation chambers **132** during part-time four-wheel drive operation, thereby providing a torque limiting feature to prevent damage to clutch pack **64**.

When mode selector **54** indicates selection of the on-demand four-wheel drive mode, ECU **56** actuates control valve **198** so as to connect third flow path **210** to sixth flow path **216** while also connecting seventh flow path **218** to fifth flow path **214**. As such, return chambers **134** are vented and supply actuation chambers **132** are supplied with pressurized fluid from EVO valve **196**. ECU **56** functions to adaptively control EVO valve **196** so as to initially supply fluid at a predetermined relatively low pressure to actuation chambers **132** that causes actuator ring **102** to index slightly relative to reaction ring **100** in the first direction. This angular movements causes actuator ring **102** to move from its low pressure position to an intermediate or "ready" position which, in turn, results in ball ramp unit **94** moving apply plate **96** from its released position to a "stand-by" position. In the stand-by position, apply plate **96** exerts a small clutch engagement force on clutch pack **64**. Accordingly, a small amount of drive torque is delivered to pinion shaft **44** through transfer clutch **46** in this adapt-ready condition. Thereafter, ECU **56** determines when and how much drive torque needs to be transferred to pinion shaft **44** based on the current tractive conditions and/or operating characteristics of the motor vehicle, as detected by sensors **52**.

Sensors **52** detect such parameters as, for example, the rotary speed of the input and pinion shafts, the vehicle speed and/or acceleration, the transmission gear, the on/off status of the brakes, the steering angle, the road conditions, etc. Such sensor signals are used by ECU **56** to determine a desired output torque value utilizing a control logic scheme that is incorporated into ECU **56**. In particular, the control logic correlates the desired torque value to a fluid pressure value to be delivered to actuation chambers **132**. Based on this desired pressure value, ECU **56** actively controls actuation of EVO valve **196** to generate a corresponding pressure level in actuation chambers **132**. Pressure sensor **202** provides ECU **56** with direct feedback as to the actual fluid pressure in actuation chambers **132** so as to permit precise control of clutch actuator **52**.

In addition to adaptive on-demand torque control, the present invention permits automatic release of transfer clutch **46** in the event of an ABS braking condition or during the occurrence of an over-temperature condition. Specifically, when ECU **56** is signaled that an ABS brake condition occurs, control valve **198** is immediately actuated to connect return chambers **134** with EVO valve **196** while actuation chambers **132** are vented to sump **192**. Also, EVO valve **196** is fully opened to send full pressure to return chambers **134**, thereby forcibly moving actuator ring **102** in its second direction to its low pressure position for fully releasing transfer clutch **46**. Moreover, if temperature sensor **200** detects that the fluid temperature in sump **192** exceeds a predetermined threshold value, the same type of immediate release of transfer clutch **46** will be commanded by ECU **56**.

While the control scheme has been described based on an on-demand torque transfer strategy, it is contemplated that a differential "mimic" control strategy can likewise be used. Specifically, the torque distribution between input shaft **42** and pinion shaft **44** can be controlled to normally maintain a predetermined rear/front ratio (i.e., 70:30, 50:50, etc.) so as to simulate the inter-axle torque splitting feature typically provided by a center differential unit. This desired torque distribution can then be adaptively controlled to address lost traction at either set of wheels. Regardless of whether the control logic scheme is based on an on-demand or a differential torque transfer strategy, accurate control of the fluid pressure delivered to actuation chambers **132** of rotary operator **92** provides the desired torque transfer characteristics being established across transfer clutch **46**.

It is contemplated that the 3-position directional control valve **198** shown in FIG. **5** could easily be substituted with a 2-position directional valve or any other known valve device capable of controlling the direction of flow in the hydraulic circuit. Likewise, FIG. **6** illustrates a modified arrangement for electrohydraulic control system **50** where control valve **198** now receives the line pressure directly from pump **190** via flow path **208** and EVO valve **196** is functional to regulate the fluid pressure within actuation chambers **132** by controlling the fluid pressure within flow path **216**. Again, EVO valve **196** is normally open with the by-passed fluid delivered through flow path **212** to cool and lubricate clutch pack **64**. In operation, the fluid pressure supplied to actuation chambers **132** is increased by closing EVO valve **196**. As such, variable control of EVO valve **156** again results in variable pressure control within actuation chambers **132**.

FIG. **7** is a hydraulic schematic of an electrohydraulic clutch system **50** which is generally similar to FIG. **6** except that control valve **198** has been eliminated. As such, ECU **56** controls the pumping direction of pump **190** by reversing the polarity of motor **194**. Check valves **230** and **232** are provided to permit such a bi-directional pumping action. As before, the pressure profile of the fluid delivered to actuation chambers **132** is adaptively controlled via variable actuation of EVO valve **196**. Optionally, EVO valve **196** can be eliminated with the pressure profile within actuation chambers **132** and return chambers **134** directly controlled by variable control of the rotary speed (RPM) and direction of electric motor **194**.

The previous hydraulic circuits utilized control valve **198** for directional control in conjunction with EVO valve **196** for adaptive pressure regulation. As an alternative, FIG. **8** illustrates a hydraulic circuit wherein a proportional control valve **234** is used to regulate both the directional and pressure characteristics. Preferably, proportional control valve **234** is a pulse width modulated (PWM) valve having a moveable element that is controlled by an electromagnetic solenoid based on electric control signals from ECU **56**. In addition, a flow control valve **236** is provided in flow path **208** to prevent dead-heading of pump **190** and to provide cooling flow to clutch pack **64**.

The arrangement shown for drive axle assembly **34** of FIG. **2** is operable to provide the on-demand four-wheel drive mode by adaptively transferring drive torque from primary driveline **14** (via propshaft **28**) to secondary driveline (via pinion shaft **44**). In contrast, a drive axle assembly **34**A is shown in FIG. **9** with torque coupling **38** now installed between differential case **70** and one of axleshafts **30** to provide an adaptive system for biasing the torque and limiting intra-axle slip between the rear wheels **32**. As before, torque coupling **38** is schematically shown to include transfer clutch **46** and clutch actuator **48**, the construction and function of which are understood to be similar to the detailed description previously provided herein for each sub-assembly. It will be understood that this particular "limited slip" differential arrangement can either be used in association with the on-demand drive axle assembly shown in FIG. **2** or in association with a drive axle assembly wherein propshaft **28** directly drives differential unit **40**.

Referring now to FIG. **10**, a drive axle assembly **34**B is schematically shown to include a pair of torque couplings **38**L and **38**R that are operably installed between a driven shaft **44** or **28** and axleshafts **30**. The driven pinion shaft drives a right-angled gearset including pinion **66** and ring gear **68** which, in turn, drives a transfer shaft **240**. First torque coupling **38**L is shown disposed between transfer shaft **240** and the left axleshafts **30** while second torque coupling **38**R is disposed between transfer shaft **240** and the right axle shaft **30**. Each coupling includes a corresponding transfer clutch **46**L, **46**R and a clutch actuator **48**L, **48**R. Accordingly, independent slip control between the driven pinion shaft and each wheel **32** is provided by this arrangement. A common sump is provided with ECU **56** controlling independent actuation of both clutch actuators **48**L and **48**R.

To illustrate an alternative power transmission device to which the present invention is applicable, FIG. **11** schematically depicts a front-wheel based four-wheel drive drivetrain layout **10'** for a motor vehicle. In particular, engine **18** drives a multi-speed transaxle **20** having an integrated front differential unit **25** for driving front wheels **22** via axle shafts **24**. As before, PTU **26** is also driven by transaxle **20** for delivering drive torque to the input member of a torque coupling **238**. The output member of torque coupling **238** is coupled to propshaft **28** which, in turn, drives rear wheels **32** via axle assembly **34**. Rear axle assembly **34** can be a traditional driven axle with a differential or, in the alternative, be similar to the arrangements described in association with FIGS. **9** or **10**. Accordingly, in response to the detection of a front wheel slip condition, torque coupling unit **238** is adaptively actuated to deliver drive torque "on-demand" to rear wheels **32**. Again, it is contemplated that torque coupling unit **238** is substantially similar in structure and function to that of torque coupling unit **38** previously described herein.

Referring now to FIG. **12**, torque coupling **238** is schematically illustrated in association with an on-demand four-wheel drive system based on a front-wheel drive vehicle similar to that shown in FIG. **11**. In particular, an output shaft **240** of transaxle **20** is shown to drive an output gear **242** which, in turn, drives an input gear **244** that is fixed to a carrier **246** associated with front differential unit **25**. To provide drive torque to front wheels **22**, front differential unit **25** includes a pair of side gears **248** that are connected to front wheels **22** via axleshafts **24**. Differential unit **25** also includes pinions **250** that are rotatably supported on pinion shafts fixed to carrier **246** and which are meshed with side gears **248**. A transfer shaft **252** is provided for transferring drive torque from carrier **246** to a clutch hub **62** associated with transfer clutch **46**. PTU **26** is a right-angled drive mechanism including a ring gear **254** fixed for rotation with drum **60** of transfer clutch **46** and which is meshed with a pinion gear **256** fixed for rotation with propshaft **28**. According to the present invention, the components schematically shown for torque transfer mechanism **238** are understood to be similar to those previously described. In operation, the power transfer system permits drive torque to be adaptively transferred from the primary (i.e., front) driveline to the secondary (i.e., rear) driveline.

Referring to FIG. **13**, a modified version of the power transmission device shown in FIG. **12** now includes a second torque coupling **238**A that is arranged to provide a limited slip feature in association with primary differential **25**. As before, torque coupling **238** provides on-demand transfer of drive torque from the primary driveline to the secondary driveline. In addition, second torque coupling **238**A now provides on-demand torque biasing (side-to-side) between axleshafts **24** of primary driveline **14**.

FIG. **14** illustrates another modified version of FIG. **12** wherein an on-demand four-wheel drive system is shown based on a rear-wheel drive motor vehicle that is arranged to normally deliver drive torque to rear wheels **32** while selectively transmitting drive torque to front wheels **22** through a torque coupling **238**. In this arrangement, drive torque is transmitted directly from transmission output shaft **240** to power transfer unit **26** via a drive shaft **260** which interconnects input gear **244** to ring gear **254**. To provide drive torque to front wheels **22**, torque coupling **238** is shown operably disposed between drive shaft **260** and transfer shaft **252**. In particular, transfer clutch **46** is arranged such that drum **60** is driven with ring gear **254** by drive shaft **260**. As such, clutch actuator **48** can be adaptively actuated to transfer drive torque from drum **60** through clutch pack **64** to hub **62** which, in turn, drives carrier **246** of front differential unit **25** via transfer shaft **252**.

In addition to the on-demand four-wheel drive systems shown previously, the power transmission technology of the present invention can likewise be used in full-time four-wheel drive systems to adaptively bias the torque distribution transmitted by a center or "interaxle" differential unit to the front and rear drivelines. For example, FIG. **15** schematically illustrates a full-time four-wheel drive system which is generally similar to the on-demand four-wheel drive system shown in FIG. **14** with the exception that an interaxle differential unit **270** is now operably installed between carrier **246** of front differential unit **25** and transfer shaft **252**. In particular, output gear **244** is fixed for rotation with a carrier **272** of interaxle differential **270** from which pinion gears **274** are rotatably supported. A first side gear **276** is meshed with pinion gears **274** and is fixed for rotation with drive shaft **260** so as to be drivingly interconnected to the rear driveline through power transfer unit **26**. Likewise, a second side gear **278** is meshed with pinion gears **274** and is fixed for rotation with carrier **246** of front differential unit **25** so as to be drivingly interconnected to the front driveline. Torque transfer mechanism **238** is now shown to be operably disposed between side gears **276** and **278**. Torque transfer mechanism **238** is operably arranged between the driven outputs of interaxle differential **270** for providing an adaptive torque biasing and slip limiting function.

Referring now to FIG. **16**, a drivetrain for a four-wheel drive vehicle is shown to include engine **18**, a multi-speed transmission **20'** for delivering drive torque to a primary or rear driveline **16'** through a power transmission device, thereinafter referred to as transfer case **300**. As seen, transfer case **300** has a rear output shaft **302** interconnected between the output of transmission **20'** and a rear propshaft **28'**. Further, propshaft **28'** is shown to drive a pinion shaft **44'** for driving a drive axle assembly which, in this example, is similar to rear axle assembly **34**A of FIG. **9**. A secondary or front driveline **14'** includes a front propshaft **304** interconnecting a front output shaft **306** of transfer case **300** to a conventional front axle assembly **308**. A transfer assembly associated with transfer case **300** includes a first sprocket **310** rotatably supported on rear output shaft **302**, a second sprocket **312** fixed to front output shaft **306**, and a chain **314** enmeshed therebetween. Transfer case **300** is shown to include a torque coupling **38** for providing on-demand transfer of drive torque from rear output shaft **302** through the transfer assembly to first output shaft **306**. As seen, transfer case **300** has an electrohydraulic control system **50'** that is controlled by ECU **56** in coordination with electrohydraulic control system **50** associated with torque coupling **38** in drive axle **34**A.

Referring now to FIG. **17**, a full-time 4 WD system is shown to include a transfer case **300'** which is generally similar to transfer case **300** of FIG. **16** except that an interaxle differential **320** is provided between an input shaft **322** and output shafts **302** and **306**. As is conventional, input shaft **322** is driven by the output of transmission **20'**. Differential **320** includes an input defined as a planet carrier **324**, a first output defined as a first sun gear **326**, a second output defined as a second sun gear **328**, and a gearset for permitting speed differentiation between first and second sun gears **326** and **328**. The gearset includes a plurality of meshed pairs of first planet gears **330** and second pinions **332** which are rotatably supported by carrier **324**. First planet gears **330** are shown to mesh with first sun gear **326** while second planet gears **332** are meshed with second sun gear **328**. First sun gear **326** is fixed for rotation with rear output shaft **302** so as to transmit drive torque to rear driveline **16'**. To transmit drive torque to front driveline **14'**, second sun gear **328** is coupled to the transfer assembly which again includes first sprocket **310** rotatably supported on rear output shaft **302**, second sprocket **312** fixed to front output shaft **306**, and power chain **314**.

While the electrohydraulic control system **50** has been described and illustrated thus far as having a clutch actuator **48** with an actuator ring **102** that is subjected to uniformly increasing fluid pressures to effect movement of the actuator ring **102** in a rotational direction that tends to increase the compressive clutch engagement force that is exerted on the clutch pack **64**, it will be appreciated that the invention, in its broadest aspects, may be constructed somewhat differently. For example, the rotary operator **92***a* may be constructed as shown in FIGS. **18** and **19**. In this example, the electrohydraulic control system **50***a* can be configured so that the torque that is produced by the clutch actuator **48***a* can be varied based on the position of the actuator ring **102***a* relative to the reaction ring **100***a*.

With reference to FIGS. **19** through **21**, the reaction ring **100***a* can be generally similar to the reaction ring **100** that is illustrated in FIG. **4**. The projecting lugs **112***a* can be configured to sealingly engage an outer portion **1000** of the hub **1002** of the actuator ring **102***a*. In the particular example provided, a seal member **1004** is housed in each projecting lug **112***a* and is configured to form a seal between the reaction ring **100***a* and the actuator ring **102***a* as will be described in detail, below. Optionally, one or more hydraulic valves **1008** can be employed to cause the hydraulic pressure that acts on one or more groups of the actuation chambers **132***a* to be phased relative to other groups of the actuation chambers **132***a* according to the relative position of the reaction ring **100***a* and the actuator ring **102***a*.

The hydraulic valves **1008** can include an associated one of the supply ports **144***a* and a fluid distribution feature **1010** that will be discussed in greater detail below. One or more of the ports **144***a* that provide pressurized fluid to the actuation chambers **132***a* can be spaced or phased so as to provide pressurized fluid an associated one of the actuation chambers **132***a* when the actuator ring **102***a* has been rotated to a predetermined location relative to the reaction ring **100***a*. For example, the ports **144***a*-**1** through **144***a*-**3** are disposed between respective lugs **112***a* and separator plates **118***a* (e.g., port **144***a*-**1** is disposed between lug **112***a*-**1** and separator plate **118***a*-**1**) when the clutch actuator **48***a* outputs a minimum compressive clutch engagement force onto the clutch pack **64***a* and each of the remaining ports **144***a* are spaced progressively farther from this relative position or datum. In the particular example provided:

- the port **144***a*-**4** is spaced by an additional first predetermined offset interval **1020**-**4**, such as 4°, relative to a predetermined datum, such as datum **1022**-**4**, to provide pressurized fluid to the activation chamber **132***a*-**4** after the actuator ring **102***a* has rotated through an angle that is about equal to the first predetermined offset interval;
- the port **144***a*-**5** is spaced by an additional second predetermined offset interval **1020**-**5**, such as 8°, relative to a predetermined datum, such as datum **1022**-**5**, to provide pressurized fluid to the activation chamber

132a-**5** after the actuator ring **102**a has rotated through an angle that is about equal to the second predetermined offset interval;

the port **144**a-**6** is spaced by an additional third predetermined offset interval **1020-6** relative to a predetermined datum, such as datum **1022-6**, to provide pressurized fluid to the activation chamber **132**a-**6** after the actuator ring **102**a has rotated through an angle that is about equal to the third predetermined offset interval; and

the port **144**a-**7** is spaced by an additional fourth predetermined offset interval **1020-7** (relative to a predetermined datum, such as datum **1022-7**), such as 16°, to provide pressurized fluid to the activation chamber **132**a-**7** after the actuator ring **102**a has rotated through an angle that is about equal to the fourth predetermined offset interval.

The actuator ring **102**a can be generally similar to that which is described above in conjunction with the embodiment of FIG. **4**, except that the separator plates **118**a can be integrally formed with the remainder of the actuator ring **102**a and the fluid distribution feature **1010**, such as a groove, can be formed in the hub **1002** of the actuator ring **102**a, and one or more of the separator plates **118**a-**4** through **118**a-**7** can be configured to selectively direct pressurized fluid from an associated supply port **144**a to an associated return port **150**a (e.g., supply port **144**a-**4** can selectively supply fluid pressure directly to return port **150**a-**4**).

Each fluid distribution feature **1010** can be configured to permit fluid communication between corresponding supply ports **144**a and return ports **150**a when the actuator ring **102**a has not rotated relative to the reaction ring **100**a by a corresponding actuating interval. The actuating interval may be any appropriate interval but in the example provided, each actuating interval corresponds to an associated one of the predetermined offset intervals (i.e., the actuating interval associated with the fluid distribution feature **1010-4** can extend over approximately 4° of the circumference of the hub **1002** of the actuator ring **102**a, the actuating interval associated with the fluid distribution feature **1010-5** can extend over approximately 8° of the circumference of the hub **1002** of the actuator ring **102**a, the actuating interval associated with the fluid distribution feature **1010-6** can extend over approximately 12° of the circumference of the hub **1002** of the actuator ring **102**a and the actuating interval associated with the fluid distribution feature **1010-7** can extend over approximately 16° of the circumference of the hub **1002** of the actuator ring **102**a in the example provided).

Accordingly, when the clutch actuator **48**a (FIG. **18**) outputs a minimum compressive clutch engagement force onto the clutch pack **64**a, the reaction ring **100**a and the actuator ring **102**a can be oriented as shown in FIG. **20**. In this condition, fluid exiting the supply ports **144**a-**4** through **144**a-**7** is directed into return chambers **134**a-**4** through **134**a-**7**, respectively, and as such, the fluid does not exert a force onto the actuator ring **102**a that would cause the actuator ring **102**a to rotate in an actuating direction (i.e., a rotational direction that tends to increase the compressive clutch engagement force that is output by the clutch actuator **48**a onto the clutch pack **64**a). Consequently, only the pressurized fluid in actuation chambers **132**a-**1** through **132**a-**3** is employed to cause the actuator ring **102**a to rotate in the actuating direction. Moreover, the fluid distribution features **1010** permit fluid communication between the actuation chambers **132**a-**4** through **132**a-**7** and return chambers **134**a-**3** through **134**a-**6**, respectively, so as to selectively permit relatively low-pressure fluid to enter or exit the actuation chambers **132**a-**4** through **132**a-**7** as the actuator ring **102**a is rotated.

When the actuator ring **102**a has rotated through an angle that exceeds the actuating angle associated with one of the fluid distribution features **1010**, the seal member **1004** that is housed in a corresponding one of the projecting lugs **112**a can sealingly engage the hub **1002** of actuator ring **102**a and thereby inhibit fluid communication between corresponding actuation chambers and return chambers (e.g., actuation chamber **132**a-**4** and return chamber **134**a-**3**). Simultaneously, a corresponding separator plate (e.g., separator plate **118**a-**4**) can be positioned relative to the supply port (e.g., supply port **144**a-**4**) so that pressurized fluid is directed into a corresponding actuation chamber (e.g., actuation chamber **132**a-**4**) rather than to the corresponding return chamber (e.g., return chamber **134**a-**4**). In this way, the amount of fluid pressure that acts on the actuator ring **102**a can be varied based upon the rotational position of the actuator ring **102**a relative to the reaction ring **100**a.

Configuration of the clutch actuator **48**a in this manner can be advantageous in that the torsional output of the clutch actuator **48**a may be tailored in a desired manner without significant changes in one or more of the flow characteristics (e.g., the mass flow rate, velocity, pressure) of the pressurized fluid.

While the invention has been described in the specification and illustrated in the drawings with reference to various embodiments, it will be understood by those of ordinary skill in the art that various changes may be made and equivalents may be substituted for elements thereof without departing from the scope of the invention as defined in the claims. Furthermore, the mixing and matching of features, elements and/or functions between various embodiments is expressly contemplated herein so that one of ordinary skill in the art would appreciate from this disclosure that features, elements and/or functions of one embodiment may be incorporated into another embodiment as appropriate, unless described otherwise, above. Moreover, many modifications may be made to adapt a particular situation or material to the teachings of the invention without departing from the essential scope thereof. Therefore, it is intended that the invention not be limited to the particular embodiment illustrated by the drawings and described in the specification as the best mode presently contemplated for carrying out this invention, but that the invention will include any embodiments falling within the foregoing description and the appended claims.

What is claimed is:

1. A power transmission device comprising:

a rotary input member adapted to receive drive torque from a source of drive torque;

a rotary output member adapted to transmit drive torque to an output device;

a torque transmission mechanism operable for transferring drive torque from said input member to said output member, said torque transmission mechanism including a friction clutch operably disposed between said input member and said output member and an electrohydraulic control system for controlling engagement of said friction clutch, said electrohydraulic control system including a clutch actuator, a fluid pump, a motor driving said fluid pump, and a controller, said clutch actuator including a rotary operator and a thrust mechanism, said rotary operator having first and second components with a plurality of actuation chambers and a plurality of return chambers, said first component being fixed for rotation with one of said input and output members and said second component adapted to rotate relative to said first component, and said thrust mechanism being operable for applying a clutch engagement force on said friction clutch in response to rotation of said second component relative to said first component, said controller being operable for controlling the fluid pressure supplied to at least one of said actuation chambers and said return chambers to thereby control rotation of said second component relative to said first component of said rotary operator;

wherein said electrohydraulic control system is configured so that said actuation chambers are divided into at least two groups, wherein a first group is configured such that said actuation chamber or actuation chambers included therein contain a relatively high pressure fluid when said second component is rotated relative to said first component in an actuating direction regardless of a relative position of said first and second components and wherein a second group is configured such that said actuation chamber or actuation chambers included therein contain a relatively low pressure fluid when said second component is positioned relative to said first component within a predetermined actuating interval and contain said relatively high pressure fluid when said second component is rotated relative to said first component in said actuation direction by an amount that exceeds said predetermined actuating interval.

2. The power transmission device of claim **1**, wherein each actuation chamber of said second group is coupled in fluid connection to a corresponding return chamber via a fluid conduit when said second component is positioned relative to said first component within said predetermined actuating interval.

3. The power transmission device of claim **2**, wherein said fluid conduit is formed between said first and second components.

4. The power transmission device of claim **3**, wherein said fluid conduit includes a passage that is formed in one of said first and second components.

5. The power transmission device of claim **4**, wherein a seal member is carried by said other one of said first and second components, said seal member being sealingly engaged to said one of said first and second components and inhibiting fluid communication between said actuation chamber of said second group and said corresponding return chamber when said second component is positioned relative to said first component by an amount that exceeds said predetermined actuating interval.

6. The power transmission device of claim **1**, wherein each of said actuation chambers is associated with a distinct supply port and said supply port of each actuation chamber in said second group is shifted relative to said supply ports of said actuation chambers of said first group by a predetermined offset interval.

7. The power transmission device of claim **6**, wherein the predetermined offset interval is about equal to the predetermined actuating interval.

8. The power transmission device of claim **1**, wherein the electrohydraulic control system is configured to include a third group of actuation chambers, the third group is configured such that said actuation chamber or actuation chambers included therein contain a relatively low pressure fluid when said second component is positioned relative to said first component within a second predetermined actuating interval and contain said relatively high pressure fluid when said second component is rotated relative to said first component in said actuation direction by an amount that exceeds said second predetermined actuating interval.

9. The power transmission device of claim **8**, wherein the second predetermined actuating interval is greater than the first predetermined actuating interval.

10. The power transmission device of claim **8**, wherein each of said actuation chambers is associated with a distinct supply port and said supply port of each actuation chamber in said third group is shifted relative to said supply ports of said actuation chambers of said first group by a predetermined offset interval.

11. The power transmission device of claim **10**, wherein the predetermined offset interval is about equal to the second predetermined actuating interval.

12. The power transmission device of claim **8**, wherein the electrohydraulic control system is configured to include a fourth group of actuation chambers, the fourth group being configured such that said actuation chamber or actuation chambers included therein contain a relatively low pressure fluid when said second component is positioned relative to said first component within a third predetermined actuating interval and contain said relatively high pressure fluid when said second component is rotated relative to said first component in said actuation direction by an amount that exceeds said third predetermined actuating interval.

13. The power transmission device of claim **12**, wherein the third predetermined actuating interval is greater than the second predetermined actuating interval.

14. The power transmission device of claim **12**, wherein each of said actuation chambers is associated with a distinct supply port and said supply port of each actuation chamber in said fourth group is shifted relative to said supply ports of said actuation chambers of said first group by a predetermined offset interval.

15. The power transmission device of claim **14**, wherein the predetermined offset interval is about equal to the third predetermined actuating interval.

16. The power transmission device of claim **12**, wherein the electrohydraulic control system is configured to include a fifth group of actuation chambers, the fifth group being configured such that said actuation chamber or actuation chambers included therein contain a relatively low pressure fluid when said second component is positioned relative to said first component within a fourth predetermined actuating interval and contain said relatively high pressure fluid when said second component is rotated relative to said first component in said actuation direction by an amount that exceeds said fourth predetermined actuating interval.

17. The power transmission device of claim **16**, wherein the fourth predetermined actuating interval is greater than the third predetermined actuating interval.

18. The power transmission device of claim **16**, wherein each of said actuation chambers is associated with a distinct supply port and said supply port of each actuation chamber in said fifth group is shifted relative to said supply ports of said actuation chambers of said first group by a predetermined offset interval.

19. The power transmission device of claim **18**, wherein the predetermined offset interval is about equal to the fourth predetermined actuating interval.

20. The power transmission device of claim **1**, wherein said first component of said rotary operator is a reaction ring having a cylindrical body segment and a plurality of radially extending first lugs which define a series of channels therebetween, wherein said second component is an actuator ring having a cylindrical body segment and a plurality of radially extending second lugs which extend into said channels so as to define a plurality of said actuation chambers and return chambers, wherein said actuation chambers are in fluid communication with an outlet of said control valve, and wherein said fluid pump is operable to draw fluid from a fluid source and deliver high pressure fluid to said control valve such that selective control of said control valve results in rotary movement of said actuator ring relative to said reaction ring.

21. The power transfer device of claim **20**, wherein said actuator ring is fixed to a drive component of said thrust mechanism such that rotation of said drive component results in translational movement of a driven component of said thrust mechanism for exerting said clutch engagement force on said friction clutch.

22. The power transfer device of claim **21**, wherein said thrust mechanism is a ball ramp unit with a first cam member as its drive component, a second cam member as its driven component, and rollers retained in cam tracks formed between said first and second cam member, and wherein said cam tracks are configured to cause translational movement of said second cam member in response to rotary movement of said first cam member for applying said clutch engagement force to said friction clutch.

23. A power transfer device for use in a motor vehicle having a powertrain and first and second drivelines, comprising:

a first shaft driven by the powertrain and adapted for connection to the first driveline;

a second shaft adapted for connection to the second driveline;

a torque transmission mechanism for transferring drive torque from said first shaft to said second shaft, said torque transmission mechanism including a friction clutch operably disposed between said first shaft and said second shaft, a clutch actuator for engaging said friction clutch and a control system for operating said clutch actuator, said clutch actuator including a fluid pump, a rotary operator and a thrust mechanism, said rotary operator including first and second components which define a plurality of actuation chambers and a plurality of return chamber that are adapted to receive pressurized fluid from said pump, said first component being fixed for rotation with one of said first and second shafts and said second component adapted to rotate relative to said first component in response to the fluid pressure in said actuation and return chambers, and said thrust mechanism is operable for applying a clutch engagement force to said friction clutch in response to rotation of said second component relative to said first component, said control system including a motor driving said pump and a controller for controlling actuation of said motor and regulating the fluid pressure supplied to at least one of said actuation chambers and said return chambers;

wherein said actuation chambers are segregated into a plurality of groups, a first one of the groups being configured such that said actuation chamber or actuation chambers included therein contains a relatively high pressure fluid when said second component is rotated relative to said first component in an actuating direction regardless of a relative position of said first and second components and wherein each of the other groups is configured such that said actuation chamber or actuation chambers included therein contain a relatively low pressure fluid when said second component is positioned relative to said first component within a corresponding actuating interval and contain said relatively high pressure fluid when said second component is rotated relative to said first component in said actuation direction by an amount that exceeds said corresponding actuating interval, and wherein each of the corresponding actuating intervals is different so that a quantity of said actuating intervals that are employed to exert a force onto said first and second components to cause said second component to be rotated relative to said first component in said actuation direction is dependent upon said relative position of said first and second components.

24. The power transfer device of claim **23**, wherein each of said actuation chambers is associated with a distinct supply port and said supply port of each actuation chamber in each of said other groups is shifted relative to said supply ports of said actuation chambers of said first group an offset interval that is unique to an associated one of said other groups.

25. The power transfer device of claim **24**, wherein the offset interval of each of said other groups is about equal to said corresponding actuating interval of said other group.

26. The power transfer device of claim **23**, wherein each actuation chamber of said other groups is coupled in fluid connection via a fluid conduit to a corresponding return chamber when said second component is positioned relative to said first component within a smallest one of said corresponding actuating intervals.

27. The power transfer device of claim **26**, wherein said fluid conduit is formed between said first and second components.

28. The power transfer device of claim **27**, wherein said fluid conduit includes a passage that is formed in one of said first and second components.

29. The power transfer device of claim **23**, wherein said first component of said rotary operator is a reaction ring having a cylindrical body segment and a plurality of radially extending first lugs which define a series of channels therebetween, wherein said second component is an actuator ring having a cylindrical body segment and a plurality of radially extending second lugs which extend into said channels so as to define a plurality of said actuation chambers and return chambers, wherein said actuation chambers are in fluid communication with an outlet of said control valve, and wherein said fluid pump is operable to draw fluid from a fluid source and deliver high pressure fluid to said control valve such that selective control of said control valve results in rotary movement of said actuator ring relative to said reaction ring.

30. The power transfer device of claim **29**, wherein said actuator ring is fixed to a drive component of said thrust mechanism such that rotation of said drive component results in translational movement of a driven component of said thrust mechanism for exerting said clutch engagement force on said friction clutch.

31. The power transfer device of claim **30**, wherein said thrust mechanism is a ball ramp unit with a first cam member as its drive component, a second cam member as its driven component, and rollers retained in cam tracks formed between said first and second cam member, and wherein said cam tracks are configured to cause translational movement of said second cam member in response to rotary movement of said first cam member for applying said clutch engagement force to said friction clutch.

32. A torque transmission mechanism for use in a motor vehicle having a powertrain and a driveline, comprising:

an input member driven by the powertrain;

an output member driving the driveline;

a clutch pack operably disposed between said input and output members;

an apply plate moveable relative to said clutch pack between a first position and a second position, said apply plate is operable in its first position to apply a minimum clutch engagement force on said clutch pack and said apply plate is operable in its second position to apply a maximum clutch engagement force on said clutch pack;

a clutch actuator for controlling movement of said apply plate between its first and second positions, said clutch actuator including a fluid pump, a rotary operator and a thrust mechanism, said rotary operator having first and second components that are coaxially arranged to define an actuation chamber and a return chamber therebetween which are adapted to receive pressurized fluid from said pump, said first component of said rotary operator is fixed for rotation with one of said input and output members and said second component is adapted to rotate relative to said first component in response to the fluid pressure in said actuation and return chambers, and said thrust mechanism is operable to move said apply plate between its first and second positions in response to rotation of said second component relative to said first component; and

a control system including a controller for regulating the fluid pressure supplied to at least one of said actuation and return chambers;

wherein said first and second components cooperate to define a plurality of hydraulic valves, each of said hydraulic valves being configured to relieve fluid pressure in an associated one of said actuation chambers when a position of said second component relative to said first component is within an associated actuating interval.

33. The power transmission device of claim **32**, wherein each associated actuating interval is different.

34. The power transmission device of claim **32**, wherein each hydraulic valve includes a hydraulic conduit that is configured to transmit fluid between one of said actuation chambers and an associated one of said return chambers when said position of said second component relative to said first component is within said associated actuating interval.

35. The power transmission device of claim **34**, wherein said fluid conduit includes a groove that is formed in one of said first and second components.

36. The power transmission device of claim **34**, wherein each hydraulic valve includes a supply port and wherein a location of at least two of said supply ports relative to a datum is staggered so that relatively high pressurize fluid is directed into said actuation chambers at different times depending on a position of said second component relative to said first component.

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